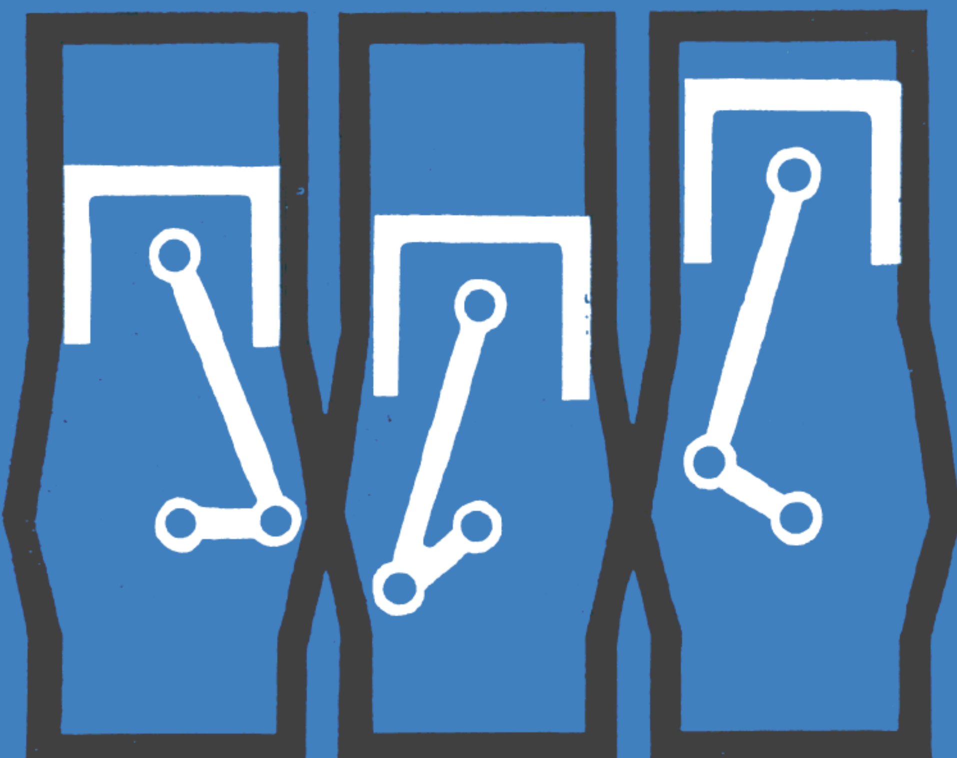
MARINE Internal Combustion Engines

A. B. KANE



MIR PUBLISHERS

MARINE Internal Combustion Engines

A. B. KANE

Translated from the Russian
by
O. K. SAPUNOV

MIR Publishers Moscow

First published 1984

Revised from the 1980 Russian edition

The Russian Alphabet and Transliteration

А а	a	К к	k	Х х	kh
Б б	b	Л л	l	Ц ц	ts
В в	v	М м	m	Ч ч	ch
Г г	g	Н н	n	Ш ш	sh
Д д	d	О о	o	Щ щ	shch
Е е	e, ye	П п	p	ъ	''
Ё ё	e, yo	Р р	r	ы	y
Ж ж	zh	С с	s	ь	'
З з	z	Т т	t	Э э	e
И и	i	У у	u	Ю ю	yu
Й й	y	Ф ф	f	Я я	ya

The Greek Alphabet

Α α	Alpha	Ι ι	Iota	Ρ ρ	Rho
Β β	Beta	Κ κ	Kappa	Σ σ	Sigma
Γ γ	Gamma	Λ λ	Lambda	Τ τ	Tau
Δ δ	Delta	Μ μ	Mu	Υ υ	Upsilon
Ε ε	Epsilon	Ν ν	Nu	Φ φ	Phi
Ζ ζ	Zeta	Ξ ξ	Xi	Χ χ	Chi
Η η	Eta	Ο ο	Omicron	Ψ ψ	Psi
Θ θ	Theta	Π π	Pi	Ω ω	Omega

На английском языке

А. Б. Кане

СУДОВЫЕ ДВИГАТЕЛИ ВНУТРЕННЕГО СГОРАНИЯ

Издательство «Судостроение»
Ленинград

MARINE INTERNAL COMBUSTION ENGINES

ИЗДАТЕЛЬСТВО МИР

CONTENTS

INTRODUCTION

The advent of internal-combustion engines relying on the heat of compression to ignite the fuel dates back to almost a century ago.

In 1893, a German engineer, Rudolf Diesel, disclosed his invention—a thermal engine designed to burn pulverized coal as fuel. However, the first model he built in 1895 appeared to be inoperative.

In 1897, having elaborated on the design, R. Diesel produced a water-cooled engine utilizing kerosene which was atomized with the aid of compressed air. Combustion occurred at a constant pressure, with air fed into the cylinder under a pressure of 294×10^4 N/m^2 (30 kgf/cm^2). The new engine developed 14.7 kW (20 hp), consuming kerosene at a rate of 327 g/kW h (240 g/hp h). However, the high cost of kerosene and obvious shortcomings of the design handicapped a wide-spread recognition of Diesel's engines. The introduction of internal combustion engines into industrial applications could be practical only in the case where they would utilize a variety of cheap fuels.

Aware of the drawbacks inherent in the engine invented by R. Diesel, the Mechanical Works of St. Petersburg (nowadays known as the Russky Diesel Works), that acquired his patent in 1897, had to redesign the engine completely. Burning crude oil at a rate of 300 g/kW h (221 g/hp h), the new Russian engine appeared to be capable of developing 18.4 kW (25 hp) in spite of being projected as a 14.7-kW (20 hp) propulsion installation.

In 1903, the Sormovo shipyard launched the world's first motor ship "Vandal" of 800 tdw, propelled by three three-cylinder engines made by the Russky Diesel Works, each rated for 106 kW (120 hp) at 240 rpm. Her propeller shafts were driven by electric motors. The 1 000 tdw twin-screw tanker "Sarmat" launched from the same shipyard in 1904 was powered by two four-cylinder engines each developing 132 kW (180 hp) at 240 rpm. To economize on weight, space and cost, "Sarmat's" propulsion system employed electric motors to run astern and diesel engines to run ahead.

In 1907, the first system of gearing a diesel engine to paddle wheels was developed at the Kolomna Works to be installed in the motor tug "Mysl'" with a 220-kW (300 hp) geared engine. An electromagnetic clutch invented by R.A. Koreivo, the Works' designer, was used for reversing and maneuvering the motor tug.

In 1908, the world's first four-stroke reversible diesel engine of the Russky Diesel Works design was built to propel the "Minoga" sub-

marine. The starting and reversing gear operated trouble-free, enabling a reversing to be completed in 10 to 12 seconds.

1910 witnessed the manufacture of diesel engines to be installed on board the gunboats "Kars" and "Ardagan", and the development of an opposed-piston engine at the Kolomna Works.

In 1911, a V-type 148-kW (200 hp), 800 rpm engine was designed and manufactured at the Russky Diesel Works. 1914 saw the production of a 37-kW (50 hp) two-stroke engine with uniflow scavenging by way of ports with valves following suit.

Early internal combustion engines were all of the air-injection type in which fuel was injected and atomized in the cylinders by means of the air compressed to 588×10^4 N/m^2 (60 kgf/cm^2) in a special compressor. Yet in 1897 a Russian engineer G.V. Trinkler developed and patented an engine that dispensed with the compressor as a means of fuel atomization. It used a valve-closed injector in conjunction with a high-pressure fuel-injection pump for the fuel delivery and atomization. Nowadays, with the advent of advanced fuel-injection equipment, the simple, dependable airless injection diesel engine has gradually gained recognition, finally ousting the air-injection engines.

A real breakthrough has occured after 1917 when a gamut of two- and four-stroke diesel engines with a power output ranging from 37 to 4 400 kW (50-6000 hp), a speed of 125 to 600 rpm, have made their appearance in this country. They have paved the way to launching a fleet of the "Kooperatsia"-class cargo-and-passenger motor ships and a fleet of the "Skumbria"-class fishing vessels in the 1930s. Simultaneously, work was initiated on programmes aimed to develop high-power and light-weight uprated diesel engines to be used on ships and aircraft.

Much headway has been made in the theory and practice of engine building in this country, any research institute, design centre and works being a beehive of activity. Far and wide are known the researchers V.A. Wanscheidt, A.S. Orlin, M.G. Kruglov, N.N. Ivanchenko who have come up with the theory of internal-combustion engines and have probed into the process of fuel injection, combustion and gas exchange in the cylinders of a diesel engine. The engines created by V.M. Yakovlev, V.A. Konstantinov, I.P. Matveev and other designers fully meet all the modern requirements. Being not inferior to foreign makes of diesels in point of performance and economy, in some aspects they compare favourably with them.

Recent development programmes are aimed at upgrading internal-combustion engines in terms of dependability, durability, economy, dimensional and weight characteristics.

Chapter I

GENERAL

1. The Concept of Prime Movers

Machines converting heat, electric, and hydraulic energy, etc. into mechanical work are prime movers. Coming under the category of thermal prime movers are steam engines, steam and gas turbines, internal-combustion engines. The working medium of a steam power plant is steam produced in a steam generator and given extra heating

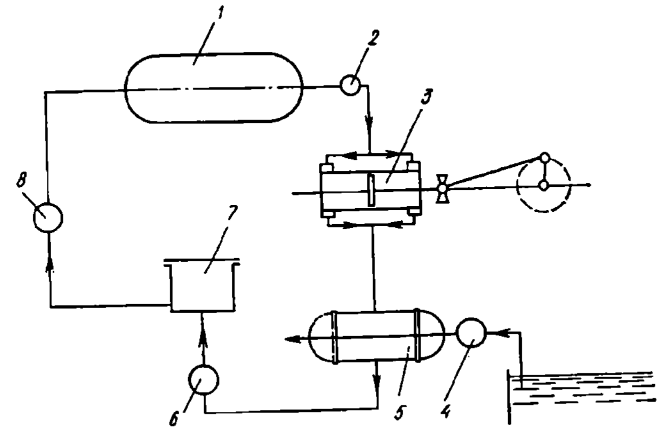


Fig. 1. Schematic diagram of converting thermal energy of fuel into mechanical work

1—steam generator; 2—superheater; 3—steam engine cylinder; 4—cooling (outboard) water pump; 5—condenser; 6—condensate pump; 7—feedwater heater; 8—feedwater pump

in a super-heater (Fig. 1). To obtain mechanical work, the steam is fed into the cylinders of a steam engine or to the blades of a steam turbine.

In a gas turbine, the working medium is gas—a product of burning a liquid fuel in a specially designed combustion chamber. Since in steam or gas turbines and in steam engines the working medium is fed from an external source, heat losses are unavoidable. On the other hand, the working medium of an internal-combustion engine is prepared directly in its cylinders, implying lower heat losses. Thus,

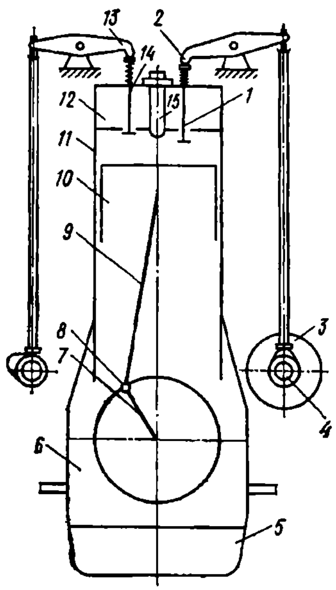


Fig. 2. Schematic diagram of internal-combustion engine

internal-combustion engines compare favourably with other types of thermal prime movers in point of economy. They are also inherently of a simpler design and more compact.

The work done in a cylinder of an internal-combustion engine by the gaseous products of combustion (the working medium) depends on displacement of the piston. This reciprocates and its motion is transformed into rotary motion of the crankshaft with the aid of the connecting rod-crank mechanism.

Referring to Fig. 2, the connecting rod-crank mechanism consists of piston *10*, connecting rod *9* and crank *7* of the crankshaft. The connecting rod-crank mechanism is contained in the engine frame the main element of which is bedplate *6*. Oil pan *5* is attached to the underside of the bedplate, and cylinder *11* with cylinder head *12* superposes the bedplate. To replace the spent working medium with a fresh one after each operating cycle, the diesel engine is provided with a valve gear comprising camshaft *4*, camshaft-timing gears *3* and *8*, inlet valve *1*, and exhaust valve *14* and rockers *2*, *13*. Fuel atomization is effected by injector *15*.

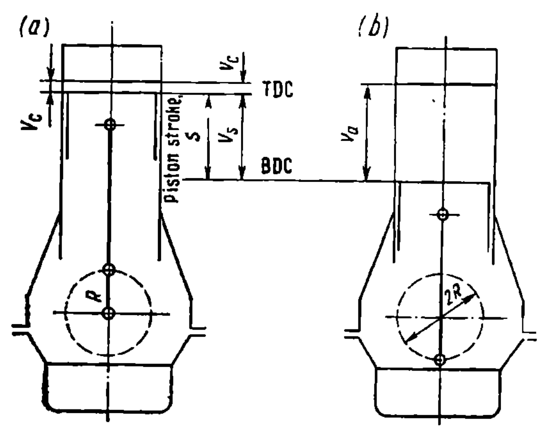


Fig. 3. Dead centres of piston: (a) top dead centre; (b) bottom dead centre

The sequence of events in the process of converting the working medium's energy into mechanical work is termed the *engine's cycle of operation*.

In a running engine, the piston moves inside the cylinder from one extremity to the other. Referring to Fig. 3, the upper extremity where the piston reverses the direction of travel and where its velocity is zero is termed the *top dead centre* (TDC); the lower extremity is called the *bottom dead centre* (BDC). The distance covered by the piston between the top and bottom dead centres is termed the *stroke* and is designated by the letter S.

The piston stroke and the crank radius are correlated, $S = 2R$, where R is the crank radius. Each piston stroke causes the crankshaft to turn through an angle of 180 deg.

The cylinder volume, V_a, a space above the piston when this is at the BDC position, equals the sum of the volume of compression chamber, V_c, and the swept volume, V_s

$$V_a = V_c + V_s$$

The swept volume is the space swept through by the piston in travelling from the TDC to BDC position

$$V_s = \frac{\pi D^2}{4} S$$

where D is the cylinder bore, cm; S is the piston stroke, cm.

The compression ratio, ε, is the ratio of the cylinder volume and the volume of the compression chamber which indicates how many times the former volume is greater than the latter one

$$\varepsilon = \frac{V_a}{V_c} = \frac{V_c + V_s}{V_c} = 1 + \frac{V_s}{V_c}$$

The mean piston speed in m/s is

$$c_m = \frac{2Sn}{60} = \frac{Sn}{30}$$

where $2S$ is the distance covered by the piston during one revolution of the crankshaft; n is the rate of crankshaft rotation.

2. Contrast between Internal-Combustion and Other Thermal Engines

Steam engines, steam and gas turbines, and internal-combustion engines can all be used in marine practice as main propulsion plants and auxiliary power units. However, unlike steam engines which are no longer installed on board the recently built ships, internal-combustion engines gain popularity on an ever-increasing scale. Their main advantages are as follows:

—the highest utilization of heat liberated by combustion and as defined by actual efficiency*, which is between 36 and 42%;

—increased ship's range without refuelling (a longer range is inherent in nuclear-powered vessels only);

—high standby efficiency with excellent maneuvering qualities;

—increased cargo-carrying capacity due to compactness of the diesel, absence of boiler room and small water requirements (the water consumption of a diesel is between 1/10 and 1/12 that of a steam engine).

However, diesel engines are not free from disadvantages which are as follows:

—inertia loads arising due to reciprocating components;

—complicated construction and high initial cost;

—continuous variations of pressure and temperature at any point in the operating cylinders;

—high idling speed of the crankshaft and its irregular rotation;

—high cost of lubricating oils (applies to medium- and high-speed diesel engines only).

3. Classification of Internal-Combustion Engines

Modern internal-combustion engines may be classified:

—by the cycle of operation, into two-stroke engines and four-stroke ones. A two-stroke engine completes its cycle of operation for every revolution of the crankshaft (or for two piston strokes). In a four-stroke engine, the cycle of operation is effected for every two revolutions of the crankshaft (or four piston strokes);

—by the mode of action, into single-acting engines and double-acting engines. A single-acting engine is one in which the cycle of operation occurs in the upper part of the cylinder, i.e. above the piston (Fig. 4*a* and *b*); in double-acting engines, the cycle of operation takes place in the upper and lower parts of the cylinder alternately. Double-acting engines have failed to gain recognition because of manufacturing and operating difficulties. Modern marine practice makes use of opposed-piston engines. Illustrated in Figs. 4*c* through *f*, in these engines each cylinder accommodates two pistons travelling oppositely to each other and forming a common combustion chamber midway of the cylinder. The power from the upper and lower pistons is transmitted to a single crankshaft in the lower part of the

* The actual efficiency is the ratio of the heat equivalent of the actual work done to the total heat liberated in the engine. It will be noted for the sake of comparison that the actual efficiency of steam engines is 12-18%, that of steam turbines is 22-32% and of gas turbines, 25-36%. The actual efficiency of marine diesels is higher because of utilizing the heat of exhaust gases and that of cooling water.

engine. Alternatively, each of the pistons operates a separate crankshaft, an upper and a lower one, the former being commonly geared to the latter which, in its turn, drives the propeller or an electric generator;

—by the fuel used, into engines run on petroleum fuel (gasoline, kerosene, naphtha, gas oil, diesel fuel), heavy fuel (motor oil, burner

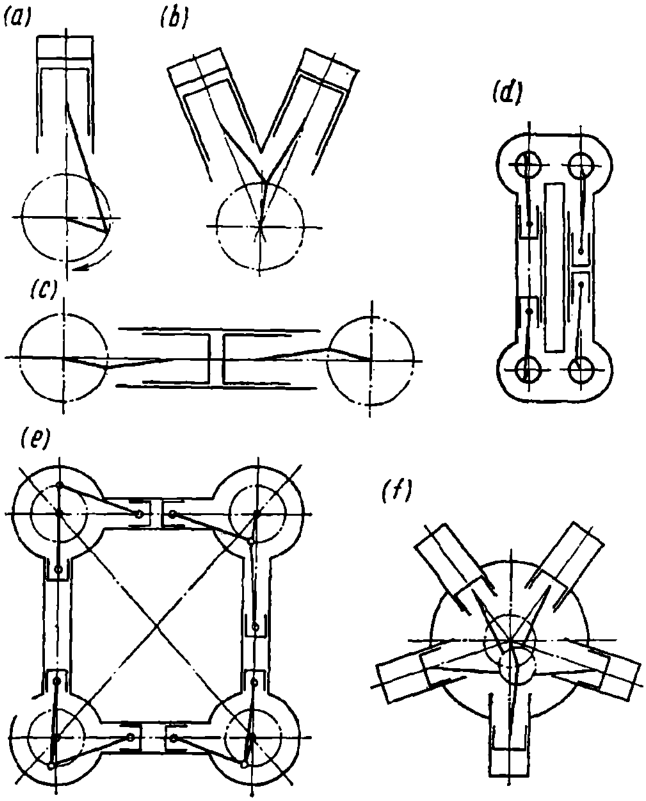


Fig. 4. Typical cylinder arrangements in marine diesel engines

fuel), gaseous fuel (natural and producer gas) and mixed fuel (engines run on gaseous fuel require a liquid fuel to initiate combustion);

—by way of charging the cylinder with the combustible mixture or fresh air, into naturally-aspirated engines and supercharged engines. In the former, air is drawn by the piston itself (applies to four-stroke engines) or is fed by a scavenging pump (applies to two-stroke engines) under a pressure exceeding the atmospheric one by

14.7×10^3 to 39.2×10^3 N/m² (0.15-0.40 kgf/cm²). In engines with supercharging, air is pressure-fed by a mechanically- or exhaust-driven blower;

—by way of igniting the combustion mixture, into compression-ignition engines (diesels) and spark-ignition engines (carburetor and gas engines). In the compression-ignition engines the injected fuel ignites the hot highly compressed air in the cylinder. In the spark-ignition engines, combustion is initiated by an electrical spark applied from an external source;

—by way of mixing air and fuel, into engines in which atomized fuel and air are mixed inside a cylinder (diesels), and engines relying on a mixture of petroleum or gaseous fuel and the air prepared outside the cylinders (carburetor and gas engines);

—by the design features, into trunk engines and crosshead engines. In the trunk engine, to keep the normal component N of the gas pressure p on the cylinder wall within an allowable limit, use is made of a trunk-type piston, i.e. one with an extended skirt functioning as a piston guide (Fig. 4*a*). In the crosshead engine, functioning as a piston guide are the shoes sliding over the crosshead guides. Modern four-stroke engines are predominantly of a trunk type and those belonging to the two-stroke category come under the crosshead type;

—by the arrangement of cylinders, into in-line engines with all the cylinders located in the same plane, and multibank engines in which the cylinders form blocks arranged either parallel to each other or in V-, W- and X-patterns (Figs. 4*b* and *f*);

—by the number of cylinders, into single-cylinder and multi-cylinder engines. Modern marine practice favours in-line diesel propulsion plants with four to twelve cylinders;

—by application, into main propulsion engines and auxiliary engines. The former are used to drive the ship's propeller (propulsor) and the latter provide power for electric generators and engine-room auxiliaries;

—by the mean piston speed, c_m, into low-speed engines (4.5 m/s $\leqslant c_m \leqslant$ 7 m/s), medium-speed engines (7 m/s $\leqslant c_m <$ 10 m/s) and high-speed engines (10 m/s $\leqslant c_m <$ 15 m/s);

—by the rate of crankshaft rotation, n, into low-rate engines (100-350 rpm), medium-rate engines (350-750 rpm) and high-rate engines (750-2 500 rpm);

—by way of changing the direction of crankshaft rotation, into direct-reversing and unidirectional engines. The former are provided with built-in reversing mechanisms used to change the direction of crankshaft rotation from "AHEAD" to "ASTERN" and vice versa. The latter operate at all times in the same direction, and a separate reversing gear unit is added to the propulsion plant if used in conjunction with a constant-pitch propeller; in the case of a variable-

pitch propeller, the ship's reversal is accomplished by reversing the pitch of the blades;

—by the direction of crankshaft rotation, into right-hand engines and left-hand engines. In the former, the crankshaft turns clockwise for running "AHEAD" and in the latter the crankshaft turns counter-clockwise, if viewed from aft.

4. Key to Designations of Home- and Foreign-Made Diesel Engines. Main Parameters of Diesel Performance

In this country, diesel engines are identified by means of model references stipulated in the appropriate USSR State Standard.

Each diesel engine type is specified by a combination of Cyrillic letters and numbers. Where, *Ч* = four-stroke engine; *Д* = two-stroke engine; *ДД* = two-stroke double-acting engine; *Р* = reversible engine; *С* = reversing gear; П = reduction gear; *К* = crosshead engine; *Н* = supercharged engine; *Г* = gaseous fuel; and ГЖ = dual-fuel engine, i.e. the one which runs both on liquid and gaseous fuels.

The numbers used before the slant line go to indicate the cylinder bore and after the slant line, the piston stroke in centimetres.

If the letter K is omitted, it indicates that the engine is of a trunk type; non-reversible engines are identified by dropping the letter P from the model reference.

Quite frequently, the number following the cylinder bore-and-piston stroke designation is used to indicate engine modification. For example, the model reference 8ДКРН 74/160-2 means that the engine is an eight-cylinder, two-stroke, crosshead, reversible and supercharged unit with a 740-mm cylinder bore and a 1 600-mm piston stroke of the second modification.

In addition to the above standard model reference, use is made of the manufacturer's designation. For example, the engine of a standard reference, such as 16ДРПН 23/2 × 30, can be designated by the manufacturer as "58" or "61", i.e. in accordance with its gross power output.

Diesel engines of foreign make are designated by the following letters;

"Sulzer Bros." (Switzerland): A = exhaust turbosupercharged engine; D = reversible; S = crosshead; R = controlled exhaust; T = trunk; B = four-stroke; P = scavenging blower; H = auxiliary; G = reversible reduction gear.

"Burmeister and Wain" (Denmark): V = two-stroke engine; W = double-acting; T = crosshead; B = exhaust turbocharged; F = reversible; M—four-stroke; S—stationary; H—auxiliary.

"MAN" (Federal Republic of Germany): G = trunk engine; K =

crosshead; Z = two-stroke; D = double-acting; V = four-stroke; C = exhaust turbosupercharged.

"Fiat" (Italy): C = crosshead engine; S = exhaust turbosupercharged; E = four-stroke; T = two-stroke trunk-type; SS = supercharged with separate exhaust from each cylinder; B = two-stroke crosshead-type.

"Stork-Werkspoor" (the Netherlands): H = airless-injection diesel; O = reversible; T = two-stroke; L = uniflow scavenging by way of valves.

The main diesel performance parameters are power, the rate of crankshaft rotation, mean effective pressure, mean piston speed and specific fuel consumption.

The standard nomenclature of engine power used in this country is:

—rated power, a continuous effective power at rated crankshaft revolutions as specified by the manufacturer without limitations on the period of continuous operation;

—overload power, a short-term power exceeding the rated one which can be used periodically during a limited time only;

—gross power output, a continuous effective power guaranteed by the manufacturer at specified crankshaft revolutions and under normal service loads.

According to power output engines are classified into low-powered units developing under 73.5 kW (100 hp); medium-powered plants whose output varies from 73.5 to 735 kW (100-1 000 hp); high-powered whose output ranges from 735 to 7 350 kW (1 000-10 000 hp); and superhigh-powered engines with an output over 7 350 kW (10 000 hp).

The rate of crankshaft rotation, n, is only a tentative criterion of the engine speed. The use of mean piston speed c_m as a criterion of engine speed is not always practical. More appropriate in this respect is the so-called *speed factor* determined by the relation $c_m n/100$; it varies between 4 and 15 for slow-speed engines and is 25 and upwards for high-speed ones.

Specific fuel consumption is a yardstick of engine economy. It is a function of mechanical and heat losses, the air-fuel mixture quality, combustion and some other factors.

5. Prospects for Low-Speed Diesel Engines

Increased international maritime traffic resulted in the construction of special-purpose vessels with a high cargo-carrying capacity. In building a ship, chief consideration is the provision of an adequate propulsion plant, for this is a factor influencing economical operation to a considerable extent.

The durable, dependable, economical and convenient low-speed two-stroke diesel engine, which has proved its merits in years of

service, is widely used to propel ships of various types, including modern roll-on/roll-off vessels (e.g. MS "Ivan Skuridin", low-speed diesels made at the Bryansk Engineering Works).

Recently, low-speed diesels have come into use as main engines both for large cargo ships and supertankers. With an output of 36 800 kW (50 000 hp), they created the prospect of furnishing a 73 500-kW (100 000 hp) twin-shaft propulsion plant.

However, low-speed diesels with loop scavenging are prone to develop troubles such as cylinder liner deformations because of non-uniform heating of the cylinders, due to the presence of many scavenging and exhaust ports, thus impairing engine dependability and uprating.

On the other hand, uniflow-scavenged engines, featuring a streamlined and, consequently, a less wasteful gas flow, create less stringent thermal conditions for the pistons and cylinders. This type is trouble-free in service, offers efficient recovery of the heat from exhaust gases and cooling water and lends itself readily to uprating by increasing the mean effective pressure. All in all, the uniflow-scavenged engine has a better showing than the loop-scavenged type.

Owing to research into thermal conditions of the operating cycle, mechanical and heat losses have been significantly reduced. This has improved the economy of low-speed diesel engines which consume now less fuel than ever before. Also the output has been boosted due to the introduction of supercharging. Low-speed diesel engines are in demand now, and there are good reasons to believe that they will enjoy wide-spread application as marine propulsion plants for 20 to 25 years to come.

6. Prospects for Medium-Speed Diesel Engines

In modern marine practice in this country and abroad there is a tendency to fit out newly built ships not only with low-speed diesels but also with medium-speed ones in conjunction with geared or diesel-electric drives. Turning out these diesels are such companies as "Pielstick" (France), "MAN" and "MAK" (Federal Republic of Germany), Sulzer Bros. (Switzerland), Burmeister and Wain (Denmark) and others.

In demand are medium-speed diesels of the trunk type which offer the following advantages:

—lower headroom of engine rooms, for the height of a medium-speed trunk diesel is only one half, or even less than that of the height of a crosshead engine;

—reduced costs of a four-stroke medium-speed geared diesel of the trunk type, ranging from 60 to 70% that of a two-stroke crosshead diesel of the same power;

—shorter periods of lying at anchor (berth) for engine maintenance

and repair, for this can be taken care of by special service centres available at all principal ports which practice the time- and labour-saving unit method of repair at high standards of workmanship. However, MAK claim that the cost of repairing a crosshead engine is 20% less than that of a medium-speed diesel of the trunk type;

—increased propulsion efficiency achieved through selecting an optimum propeller speed;

—the possibility of pooling the output of several diesels in one propulsion plant with an aggregate power output ranging from 36.8×10^3 to 51.5×10^3 kW (50 000-70 000 hp) in the case of a single-shaft installation, and over 73 500 kW (100 000 hp) for a twin-shaft one.

On the other hand, medium-speed diesels are not free from such drawbacks as:

—higher than in low-speed engines vulnerability to defects resulting from additional valves, fuel pumps, fuel injectors, etc.;

—loss of some power in the reduction gear;

—higher noise and vibration level.

However, these drawbacks are likely to be minimized due to achievements in modern chemistry, metallurgy and mechanical engineering. So, for example, widely practiced in shipbuilding is the cowling of the engine and its auxiliaries. Sound-insulated propulsion control rooms are provided on ships equipped with medium-speed diesels which prevent the personnel from exposure to noises in excess of the permissible level. High-alloy metals and a variety of coatings used in manufacturing medium-speed diesels significantly improve wear resistance in engine components.

The use of medium-speed diesels is justified in many cases in spite of adding such items as reduction gears, couplings and pitch-controlling mechanisms to the propulsion plant.

REVIEW QUESTIONS

1. How can energy be converted from one form into another?
2. What is the difference between internal-combustion engines and other thermal prime movers?
3. How are marine internal-combustion engines classified?
4. What are the main performance parameters of a diesel engine?
5. What are the prospects for marine internal-combustion engines?

Chapter II

FUNDAMENTALS OF INTERNAL-COMBUSTION ENGINES

7. Operating Cycle of a Four-Stroke Diesel Engine

A four-stroke engine is one in which the operating cycle is completed in four strokes of the piston. Depending on the direction of piston travel and the process taking place in the cylinder during this travel, the strokes are referred to as suction (induction), compression, working (expansion) and exhaust strokes.

Suction is the first stroke (Fig. 5*a*). It begins with the departure of the piston, due to the rotation of the crankshaft, from the TDC position so that a vacuum is created in the space above the piston. Urged by the pressure difference, atmospheric air fills the cylinder through an open inlet valve as long as the piston travels towards the BDC position. The admission of air terminates when the inlet valve closes, the pressure in the cylinder being 83.5×10^3-93.5×10^3 N/m^2 (0.85-0.95 kgf/cm^2) and the temperature, 310-320 K (37-47°C).

Compression is the second stroke (Fig. 5*b*). The piston moves from BDC to TDC due to the crankshaft action, compressing the fresh air charge to 294×10^3-442×10^3 N/m^2 (30-45 kgf/cm^2) with an increase in the temperature to 873-973 K (600-700°C), while the inlet and outlet valves stay closed. Towards the end of the compression stroke, fuel is injected into the cylinder under a pressure of from 196×10^5 to 1470×10^5 N/m^2 (200-1 500 kgf/cm^2) depending on the diesel type. The finely atomized fuel particles intermix with the air to form a combustible mixture which, on going through some physical and chemical changes, self-ignites due to the high temperature of the compressed air in the cylinder.

Theoretically, the fuel injection must be precisely timed to occur at an instant between 10 and 35 degrees of the crankshaft rotation corresponding to a certain distance of the piston from the TDC position. Practically, this instant must be selected within a given interval, termed injection advance, so as to coincide with the peak of temperature in the cylinder, ensuring thereby optimum conditions for the combustion. Thus, the stages of the second stroke are compression of the air, intermixture of the air and fuel charges, and initiation of the combustion. The pressure and temperature of the gases in the cylinder at the end of the combustion are 58.2×10^5-75.8×10^5 N/m^2 (60-80 kgf/cm^2) and 1 873-2 273 K (1 600-2 000°C), respectively.

Expansion is the third stroke (Fig. 5*c*). Moving towards BDC due to the high gas pressure, the piston applies force to the crankpin through the connecting rod, converting rectilinear movement into

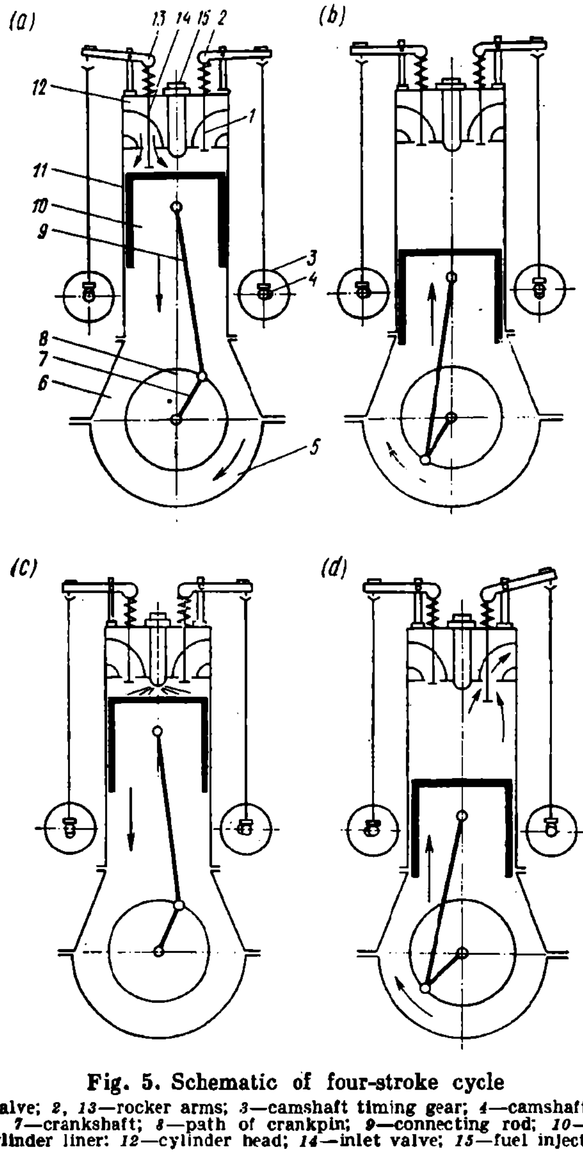


Fig. 5. Schematic of four-stroke cycle

1—exhaust valve; *2*, *13*—rocker arms; *3*—camshaft timing gear; *4*—camshaft; *5*—oil pan; *6*—crankcase; *7*—crankshaft; *8*—path of crankpin; *9*—connecting rod; *10*—piston; *11*—cylinder liner: *12*—cylinder head; *14*—inlet valve; *15*—fuel injector

rotation. This is why the third stroke is termed the *working stroke*. The pressure and the temperature decrease to 34.3×10^4-49×10^4 N/m^2 (3.5-5 kgf/cm^2) and 1 023-1 173 K (750-900°C), respectively, due to the expansion of space above the piston.

Exhaust is the fourth stroke (Fig. 5*d*). It begins with the opening of the exhaust valve when the piston approaches BDC through which the gases having a pressure exceeding the atmospheric one leave into the exhaust manifold. The rest of gases are expelled from

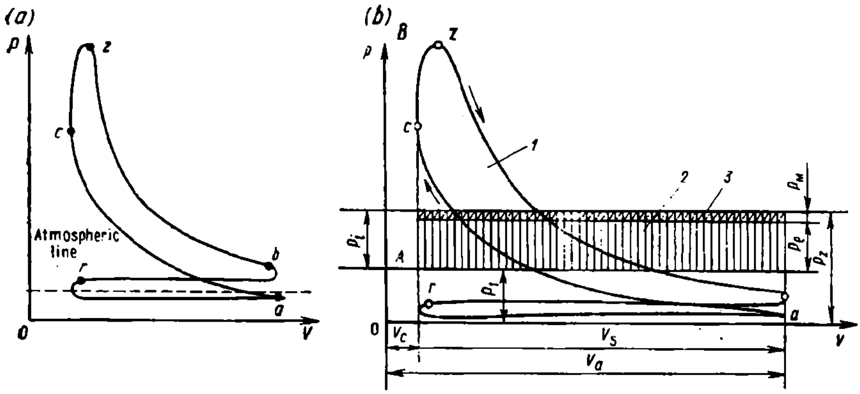


Fig. 6. Indicator diagram of four-stroke diesel engine

the cylinder by the piston as it reverses its travel and begins to move towards TDC. The exhaust pressure is 10.8×10^4-11.8×10^4 N/m^2 (1.1-1.2 kgf/cm^2) and the corresponding temperature is 703-803 K (430-530°C).

The cycle of operation comes to an end with the closure of the exhaust valve which is an event timed to occur after TDC.

Pressure variations inside the cylinder of a diesel engine depending on the piston position are represented by a closed curve referred to as *indicator diagram* which is plotted on the p-V (pressure versus volume) coordinates by means of a device termed indicator when the engine is in operation.

Figure 6*a* shows how each line on the diagram corresponds to a certain stage of the cycle of operation. So, the suction stroke is depicted by line *ra*, the compression stroke is represented by line *ac*, the working (firing) stroke is illustrated by line *zb*, the exhaust stroke goes on as indicated by line *br*, and the combustion of fuel is presented by line *cz*. The horizontal dotted line indicates atmospheric pressure.

Area *1* or the equivalent area *2* determines the effective engine power (Fig. 6*b*); area *3* represents mechanical losses.

From an analysis of the four-stroke diesel engine operation it can be concluded that three out of the four strokes in the cycle of operation are auxiliary strokes designed to support the working stroke. Extra energy required to cause the auxiliary strokes—suction, compression and exhaust—is supplied externally either from the working strokes of the other cylinders or from the kinetic energy stored in the flywheel.

8. The Cycle of Operation in a Two-Stroke Diesel Engine

Referred to as a two-stroke engine is one in which the operating cycle is completed in two piston strokes corresponding to one crankshaft revolution.

In this type of engine, the lower part of the cylinder liner is pierced with scavenging ports and exhaust ports grouped in the diametrically opposite halves of the liner circumference. The exhaust ports are as a rule somewhat higher than the scavenging ones. Unlike a four-stroke diesel in which suction and exhaust are separate events, a two-stroke engine combines them with other strokes of the operating cycle. So, the scavenging and filling of the cylinder with air followed by the compression of the air take place in the course of piston traveling upwards, i.e. from BDC to TDC, and the working stroke occurring during the downward piston displacement is succeeded by the exhaust into the atmosphere.

In a two-stroke diesel engine, air is pressure-fed with the aid of a crankshaft-actuated scavenging pump of a reciprocating type. Drawing in atmospheric air, the pump compresses it to some extent before delivering into a receiver (a fresh air container) and then into the cylinder by way of the scavenging ports. The burnt gases are driven out through the open exhaust valves connected to the exhaust manifold. Both the exhaust and scavenging ports are covered and uncovered by the piston as it reciprocates in the cylinder.

Consider the operating cycle of a two-stroke diesel. The first stroke covers both scavenging and the compression of air (Fig. 7*a* and *b*). When the scavenging and exhaust ports are uncovered due to the piston being at BDC, the scavenging pump feeds air into the receiver and cylinder at a pressure of 11×10^4-13.6×10^4 N/m^2 (1.15-1.40 kgf/cm^2). Filling the space above the piston, this air clears the cylinder of the burnt gases left over from the preceding cycle. As the piston sets out on its upward travel, it covers the scavenging and exhaust ports, compressing the air. At the end of the upstroke, the pressure in the cylinder builds up to 31.4×10^5-44.2×10^5 N/m^2 (32-45 kgf/cm^2) and the temperature rises to 923-1 073 K (650-800°C) which is sufficiently high to ignite the fuel.

Summing up, the first piston stroke, or one half of crankshaft

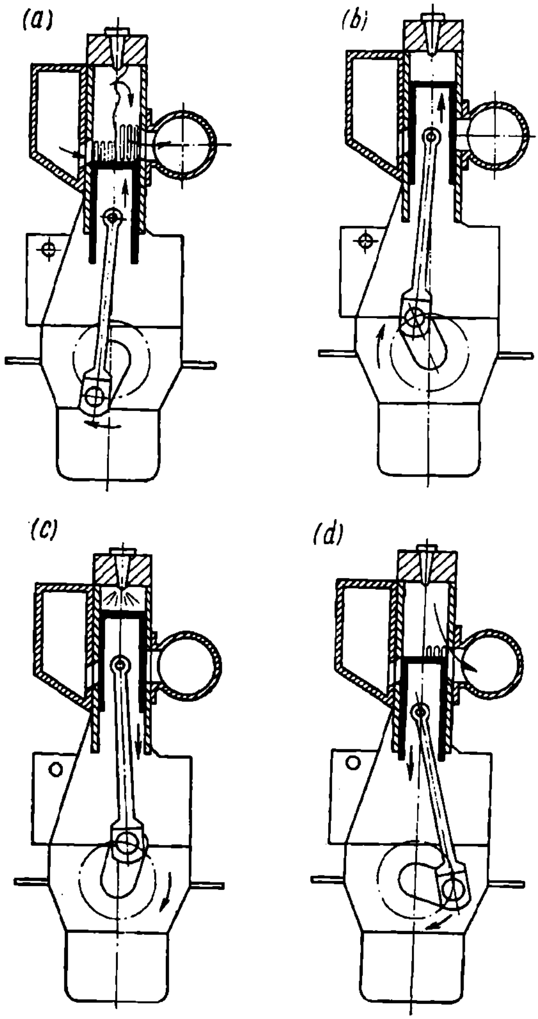


Fig. 7. Schematic of two-stroke cycle

(*a*) piston at bottom dead centre; (*b*) upstroke; (*c*) piston at top dead centre; (*d*) piston nearing bottom dead centre

revolution, initiates two processes in succession: scavenging or the admission of air and the compression of the air.

On fuel delivery through the injector the charge is self-ignited and fired. The products of combustion expand, causing the piston to move towards BDC (Fig. 7*c* and *d*).

The indicator diagram shows the following points (Fig. 8): *1* and *6*, the scavenging ports become covered; *6* and *2*, the exhaust ports become covered; *3*, advanced feeding of fuel.

Travelling downwards due to the gas pressure the piston is doing work. The uncovering of the exhaust ports by the piston, when this is between 40 and 60 deg of crankshaft rotation just before the BDC position, initiates the exhaust of the gases into the atmosphere. Consequently, the gas pressure decreases to 24.5×10^4-39.2×10^4 N/m^2 (2.5-4 kgf/cm^2), but the exhaust temperature is as high as 523-673 K (250-400°C). The exhaust ports stay uncovered during an interval of 118-130 deg of crankshaft rotation and the corresponding period for the scavenging ports is 100-140 deg on the crankshaft. It will be noted that the effective piston stroke is shorter than the geometrical one, the height of the exhaust ports making all the difference.

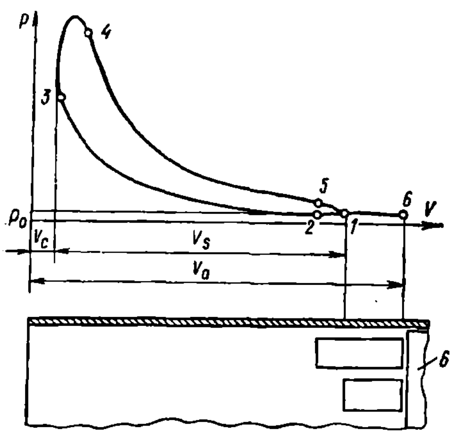


Fig. 8. Indicator diagram of two-stroke diesel engine

Thus, the second piston stroke, or the next one-half of crankshaft revolution, covers the expansion and exhaust strokes. Referring to the indicator diagram, the expansion stroke is between points *4* and *5*, the uncovering of the exhaust ports, point *5*.

Two-stroke engines appear in a multitude of designs among which of particular interest is the opposed-piston type illustrated in Fig. 4*c*. Its salient feature are two pistons working in the same cylinder. Their crowns form a combustion chamber at the points of the closest approach to each other. Scavenging ports are provided in the upper portion of the cylinder linear and their functioning is controlled by the upper piston. Exhaust ports are located in the lower portion, being covered and uncovered by the lower piston.

Each of the pistons imparts rotary motion to the upper and lower crankshafts geared to each other.

As the pistons set out travelling towards each other, they cover

the exhaust and scavenging ports and compress the air in the combustion chamber. The resulting rise in the temperature of the air ignites the fuel injected into this chamber. The gases formed due to the fuel combustion exert a pressure on the pistons, causing them to reverse the direction of travel. At the same time, the forces set up by the pressure are transmitted to the crankshafts via connecting rods. The exhaust ports uncovering ahead of the scavenging ones due to the upper crankshaft being arranged to rotate somewhat in advance of its lower counterpart reduce the pressure in the cylinders to a point where it equals the pressure in the receiver. This promotes the filling of the cylinder with fresh air at the instant the scavenging ports become uncovered.

9. The Cycle of Operation in a Four-Stroke Carburetor Engine

Carburetor engines are mostly of the four-stroke type. Unlike diesels, they use volatile fuels (gasoline or kerosene) and employ a special device termed carburetor to furnish a combustible mixture consisting of air and gasoline vapour. Ignition is initiated by an

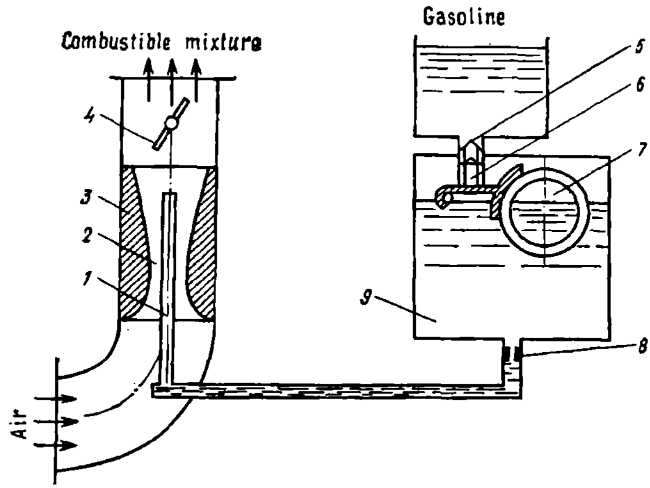


Fig. 9. Carburetor

electric discharge and not by the high-temperature air as in diesels.

A combustible mixture is said to be normal if the amount of air fed to burn 1 kg of gasoline is 15 kg. A smaller amount of air renders the mixture rich and a greater one, lean. Rich mixture is used during the starting of an engine. Continuous operation on a rich mixture is intolerable, for, apart from excessive fuel consumption, this leads

to a heavy carbonization and over-heating of the cylinders and pistons.

Correct air-fuel intermixture is performed by the carburetor (Fig. 9), and the process of preparing the combustible mixture from gasoline vapour and air is termed carburation. It is accomplished as follows.

When the piston is on the suction stroke, air enters a pipe having venturi *3* of a reduced cross section to accelerate the air flow to 120-180 m/s that creates a low-pressure area in mixing chamber *2*. A fuel jet *1* located in the venturi forms communicating vessels with float chamber *9*. The admission of fuel into the jet is controlled by calibrated orifice *8*. Due to the pressure in the mixing chamber being lower than that in the float chamber the fuel issuing from the jet is atomized by the high-velocity air stream.

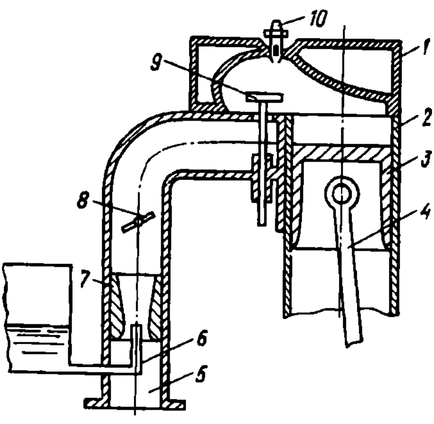


Fig. 10. Schematic of carburetor in operation

1—cylinder head; *2*—engine cylinder; *3*—piston; *4*—connecting rod; *5*—air intake; *6*—jet; *7*—venturi; *8*—throttle valve; *9*—inlet valve; *10*—spark plug

Gasoline is admitted into the float chamber from service tank through orifice *5* in the float chamber cover. Valve *6* controls the fuel inflow, being attached to float *7* which—pivoting about a horizontal pin—rises and sinks with the fuel level. This keeps the fuel in the float chamber at a constant level.

Depending on the engine load, the flow rate of the combustible mixture is controlled by a throttle valve *4*. For example, in an idling engine the throttle valve is nearly closed and the amount of mixture reaching the cylinder is small. To increase the engine output and the crankshaft speed, the opening of the throttle valve is increased so as to admit more mixture into the cylinder.

Consider the operation of a carburetor engine (Fig. 10). The combustible mixture which has been admitted into the cylinder is compressed to between 54×10^4 and 118×10^4 N/m^2 (5.5-12 kgf/cm^2), its temperature increasing to 623-673 K (350-400°C). However, this temperature is too low to effect self-ignition. The alternative is the use of a spark plug to produce an electric spark at an instant anywhere between 25 and 35 deg of a crankshaft rotation before the piston is at TDC. The combustion appears to be an almost constant-pressure process at 294×10^4-490×10^4 N/m^2 (30-50 kgf/cm^2),

giving rise to a temperature of 2 273-2 673 K (2 000-2 400°C).
High compression ratios should be carefully avoided in carburetor engines or otherwise a prohibitively high temperature in the cylinder may cause the combustible mixture to detonate and burn down explosively. For normal operation, a compression ratio of 4.5-9.5 is required. In the case of a higher compression ratio, a high-octane gasoline must be used.

In spite of such assets of carburetor engines as compactness and light weight, they use expensive and fire-hazardous fuel. The actual efficiency of carburetor engines is only 22-24% and the specific fuel consumption is much higher than in diesel engines.

10. Two-Stroke Engines Versus Four-Stroke Ones

A comparison of the operating cycles of the two-stroke and four-stroke engines makes sense only then when the cylinders of both engines are of the same geometrical dimensions and both of them are run at the same crankshaft speed.

Theoretically, the comparison reveals that the output of a two-stroke engine is twice that of a four-stroke one. This is due to the fact that in the two-stroke engine, two operating strokes occur with every two revolutions of the crankshaft, whereas in the four-stroke engine there is only one operating stroke in the same period.

In fact, the output of a two-stroke engine is 1.5 to 1.8 times that of a four-stroke unit, other things being equal. The point is that in the two-stroke engine the actual operating cycle is only a fraction of the total piston stroke, lasting between TDC and the instant of uncovering the exhaust ports. Moreover, some 8 to 11% of the indicated power are wasted in the two-stroke engine to drive the scavenging pump which clears the cylinders from the burnt gases and fills them with a fresh air charge.

It is also necessary to point out that the period available in the two-stroke engine for expelling the products of combustion from the cylinder and filling it with fresh air is much shorter than in the four-stroke engine where more than two piston strokes are allotted to that end. This means that some of the gases are left behind in the cylinder, interfering with the normal course of the operating cycle. As a result, two-stroke engines appear to be less economical in most cases than the four-stroke units.

At the beginning of compression, the two-stroke engine sustains higher pressures and temperatures than a four-stroke one does. The same applies to combustion pressure and the mean temperature of the operating cycle. So, in two-stroke engines this temperature is 1 123 K (850°C) against 873 K (600°C) in four-stroke ones, i.e., higher by 25-30%. The elevated temperature brings about a 30 to 40%

increase in the thermal load on the cylinders, which sets up high thermal stresses on the combustion chamber walls.

For trouble-free operation, the cylinder heads and piston crowns of two-stroke engines must be carefully designed and made from a material meeting stringent requirements. Operating under difficult conditions piston rings are another source of concern because of their gumming.

On the other hand, the cylinder heads of two-stroke engines employing loop scavenging need neither inlet and exhaust valves, nor a complex valve-actuating gear, and are therefore of a simple construction. The same applies to the two-stroke engine as a whole. The torque produced by a two-stroke engine is less irregular than this is the case for a four-stroke engine, because the number of operating cycles in the former is twice that in the latter, provided both engines have the same number of cylinders and operate at the same speed. The force applied to a piston of a two-stroke engine coincides with the axis of the connecting rod at all times and never changes its direction throughout the operating cycle. This eliminates dynamic loads coming on the piston crowns which are unavoidable in four-stroke engines.

The advantages of two-stroke engines become particularly pronounced, if compared with four-stroke engines, at outputs beginning with 1 100 kW (1 500 hp) and upwards and low rates of crankshaft rotation. In this case, scavenging is highly desirable in order to lessen the effect of the drawbacks inherent in the two-stroke cycle, even if the scavenging air pressure is low, anywhere between 0.98×10^4 and 1.47×10^4 N/m^2 (0.1-0.15 kgf/cm^2).

This accounts for the fact that low-speed high-powered diesel engines are widely used in marine application, successfully competing with four-stroke power plants in many respects, dependability and economy included. However, for a high-speed propulsion installation, the four-stroke cycle is practical.

11. Timing of Suction and Exhaust, Fuel Injection Advance

The inlet valve opening by the time a fresh air charge is to be admitted into the cylinder provides for the air a passage of adequate cross section and low resistance. The filling of the cylinder with air is also facilitated by the inertia of the head of inrushing air which comes into play because of the closing of the inlet valve after the piston has passed BDC, i.e. with a delay. Commonly, expressed in terms of crankshaft angle, this delay or lag is 20 to 45 deg, the higher values being ascribed to high-speed propulsion installations. Note, that the suction stroke lasts between 220 and 240 deg of a crankshaft rotation.

At the same time, to reduce the work done by the piston in overcoming the backpressure, while expelling the burnt gases from the cylinder, the exhaust valve is opened before the piston arrives at BDC, i.e. with an advance. The corresponding angle of advance is 18-45 deg of a crankshaft rotation. The exhaust valve, when open, enables the gases to escape into the exhaust manifold. Their pressure, consequently, drops to 11.8×10^4 N/m^2 (1.2 kgf/cm^2) and so does their temperature, becoming 673-773 K (400-500°C) instead of

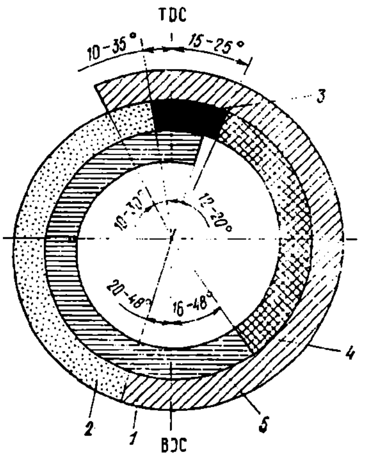


Fig. 11. Valve timing diagram of four-stroke diesel engine

1—suction stroke; 2—compression stroke; 3—fuel injection; 4—expansion stroke; 5—exhaust stroke

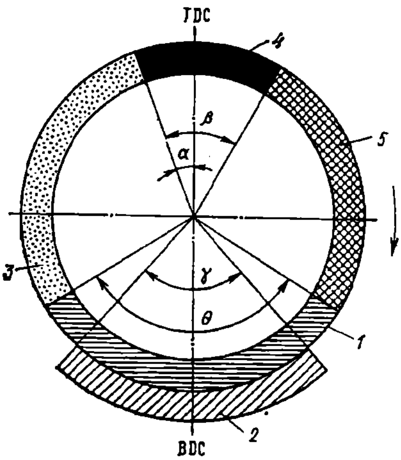


Fig. 12 Valve timing diagram of two-stroke diesel engine

1—exhaust (exhaust valves are uncovered); 2—scavenging (scavenge ports are uncovered); 3—compression; 4—fuel injection; 5—expansion

973-1 023 K (700-750°C) shortly before. The exhaust valve, in its turn, is timed to close after the piston has passed TDC, i.e. with a lag amounting to 12-20 deg of a crankshaft rotation. This provides for a better clearing of the cylinder from the gases.

The optimum angles at which valves open and close and the fuel injection into the cylinder are determined experimentally during the tests of the prototype engine on the manufacturer's test stand and are entered into the engine log. The intervals during which the inlet and exhaust valves stay open and the ports in the cylinder walls are uncovered are referred to as valve timing.

Graphically, the valve timing is represented by a circular diagram depicted in Figs. 11 and 12. It can be seen that there are intervals when both the inlet and exhaust valves are open. This condition is

referred to as valve overlapping and it may last 25-55 deg of a crankshaft rotation.

Before self-ignition diesel fuel must be heated up and vapourized, which involves certain physical and chemical changes. This all occurs not instantaneously but after some interval termed ignition lag. To allow for this lag, fuel is injected before the piston arrives at TDC, i.e. with an advance known as injection advance. Varying between 5 and 35 deg of a crankshaft rotation beginning from the instant of fuel injection into the cylinder, the injection advance is determined by the fuel system used, engine speed and fuel grade, the higher values being ascribed to high-speed diesels.

An optimum injection advance, as determined under the conditions of rated engine output, ensures good performance and economy of the propulsion plant. However, its economy will be impaired if the output is changed. To avoid this, some of the high-speed diesel engines are provided with means of adjusting the injection advance in operation depending on engine speed—the higher the speed the greater the angle of injection advance. A correctly-set injection advance provides for initiating the fuel ignition before the piston arrives at TDC. A too small injection advance results in an overlapping of the expansion stroke by the combustion which invites the overheating of cylinders, overstressing of pistons, excessive heat losses and poor economy.

The timing diagram of a two-stroke diesel engine distinctly indicates the instants when the piston uncovers and covers the scavenging and exhaust ports, the average duration of scavenging and exhaust being 75-125 and 110-145 deg of a crankshaft rotation, respectively.

12. Ideal Cycles of Internal-Combustion Engines

To obtain a clear-cut picture of the heat-exchange processes taking place in the cylinders of internal-combustion engines and to evaluate them in point of effectiveness, use should be made of the so-called ideal cycle. However, there is hardly a thermal cycle which can be adopted as the ideal one for all the internal-combustion engines because the process of combustion in them is not the same. In fact, the combustion in a carburetor engine is an almost constant-volume process, and in the air-injection diesel engine, combustion is a constnt-pressure process while in the airless-injection diesel engine it is the combination of the two processes.

Consider the thermodynamic cycle of internal-combustion engines on the following assumptions. In the ideal cycle, the compression and expansion are adiabatic processes, i.e. processes during which no heat transfer takes place between the gas in the cylinder and the surroundings (in an actual cycle, intensive heat transfer is observed during the compression and expansion). Moreover, in the ideal cycle

the combustion of fuel is regarded as an input of heat from an external source without any changes in the chemical composition of the working medium, and the rejection of heat to a low-temperature medium (exhaust) is regarded as a constant-volume process. The specific heat capacity is assumed to be constant during the processes which are not regarded to be influenced by temperature.

By virtue of these assumptions, it is possible to use simple mathematical relations and consider factors influencing the economy and effectiveness of the cycle. Thus, the ideal thermodynamic cycles are treated as prototypes of the actual processes taking place in internal-combustion engines.

In a thermodynamic cycle heat can be supplied or rejected at one or more stages either with or without changes in the volume of the working medium. Depicted in Fig. 13*a* is an ideal cycle in which inputs of heat occur at constant volume and constant pressure. Accordingly, *ac* represents adiabatic compression, *cz'* and *ab* are isochores defining constant-volume processes, *z'z* is an isobar (a constant-pressure process) and *zb* illustrates adiabatic expansion. Heat is supplied within the segments *cz'* and *z'z* and rejected along *ba*.

Let us assume that the cycle takes place in an ideal engine, and the cylinder size of this engine is equal to the cylinder size of the actual engine. Then, by analogy with the actual engine:

$$V_s = \text{swept volume of the cylinder}$$

$$V_c = \text{volume of the compression chamber}$$

$$V_a = V_s + V_c = \text{cylinder volume}$$

$$\varepsilon = \frac{V_a}{V_c} = \text{compression ratio}$$

$$\rho = \frac{V_z}{V_c} = \text{ratio of constant-pressure expansion}$$

$$\delta = \frac{V_b}{V_z} = \frac{V_a}{V_z} = \text{ratio of adiabatic expansion}$$

$$\lambda = \frac{p_z}{p_c} = \text{combustion pressure ratio}$$

For diesel engines, $\rho = 1.4\text{-}1.7$; $\delta = 8\text{-}9.5$; $\lambda = 1.3\text{-}2.5$.

The economy of a cycle depends on the so-called *thermal efficiency* which is the ratio of the heat equivalent of the work done to the total heat input. The thermal efficiency of any particular cycle can be determined as follows.

1. For a cycle with a constant-volume heat input (Fig. 13*b*):

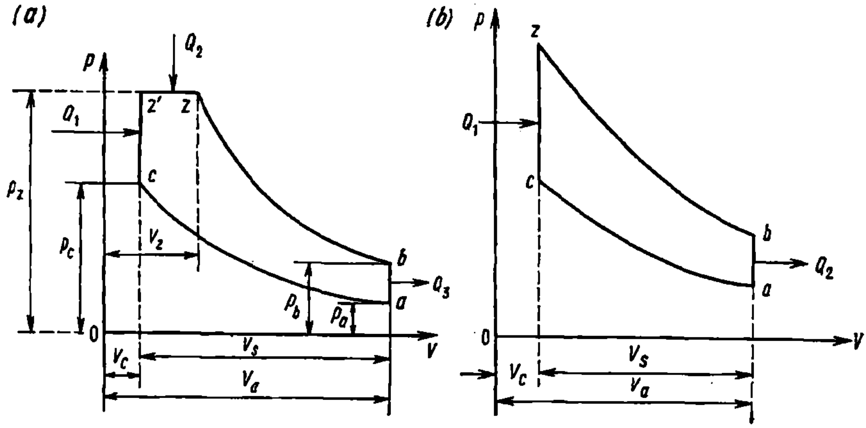
$$\eta_t = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1}$$

where Q_1 = heat supplied to the working medium during the cycle; Q_2 = heat rejected to a low-temperature body.

The useful heat, i.e. one converted into work is

$$Q_u = Q_1 - Q_2$$

Before considering thermal efficiency of the cycle in detail, it is appropriate to determine the temperatures at each representative



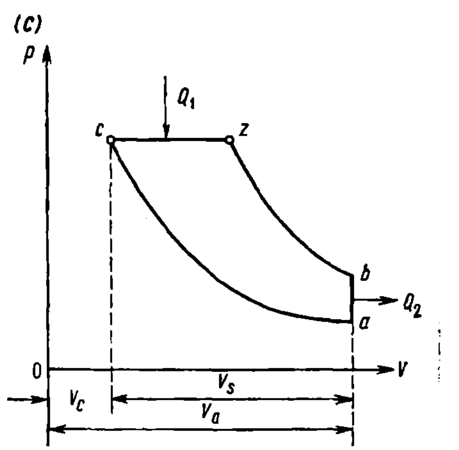


Fig. 13. Ideal cycles of diesel engine

point of this cycle, using relationships known from thermodynamics.

So, the temperature at the end of the compression is $T_c = T_a \varepsilon^{k-1}$, where $k = c_p/c_v$ = exponent of the adiabatic curve; c_p and c_v = specific heat capacities of gas at constant pressure and constant volume, respectively. The quantities of heat supplied and rejected in this case

$$Q_1 = c_v (T_z - T_c) \quad \text{and} \quad Q_2 = c_v (T_b - T_a)$$

The maximum temperature at the end of the constant-volume heat input

$$T_z = T_a \varepsilon^{k-1} \lambda$$

The temperature at the end of the expansion of the working medium is given by

$$T_b = T_a \lambda$$

Substituting the above values of temperatures into the equation of the thermal efficien y of the cycle, we obtain

$$\eta_t = 1 - \frac{c_v (T_b - T_a)}{c_v (T_z - T_c)} = 1 - \frac{T_b - T_a}{T_z - T_c}$$

$$= 1 - \frac{T_a \lambda - T_a}{T_a \varepsilon^{k-1} \lambda - T_a \varepsilon^{k-1}}$$

$$= 1 - \frac{T_a (\lambda - 1)}{T_a \varepsilon^{k-1} (\lambda - 1)} = 1 - \frac{1}{\varepsilon^{k-1}}$$

2. For a cycle with a constant-pressure heat input (Fig. 13*c*), the thermal efficiency is determined by analogy with the preceding cycle, i.e. by substituting the values of the temperatures at the appropriate points of the cycle into the equation of thermal efficiency, with p = const

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \frac{\rho^k - 1}{k (\rho - 1)}$$

3. For a combined cycle, i.e. one with a constant-volume heat input and a constant-pressure heat input (Fig. 13*a*)

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \frac{\lambda \rho^k - 1}{\lambda - 1 + k\lambda (\rho - 1)}$$

The total amount of heat added during the combined cycle may be distributed between the V = const and p = const stages in various ways, which would influence the values of λ and ρ.

Comparing the ideal cycles with each other, we come to the following conclusions:

1. For V = const, the thermal efficiency of the cycle is a function of the compression ratio ε and the exponent k.

2. For p = const, the thermal efficiency rises with the compression ratio ε and decreases with an increase in the ratio of constant-pressure expansion ρ. High compression ratios are conducive to improved economy of the cycle.

3. For V = const and p = const, an increase in the compression ratio ε leads to improved economy of the cycle which stands between the cycles of the constant-volu . e and constant-pressure heat input. An increase in the combustion pressure ratio λ results in a better utilization of the heat during the cycle.

4. Ranking highly in point of economy are the airless-injection diesel engines which are capable of ensuring a maximum compression ratio ε. Their theoretical basis is the combined cycle.

13. Specifics of the Working Cycle of the Internal-Combustion Engine

The actual or working cycle of the engine involves thermal, hydraulic and mechanical losses which cannot be taken into account by any of the ideal cycles.

A fresh air charge filling a cylinder thermally interacts with the hot cylinder walls and the hot gases. Some of the energy produced by the engine is spent to overcome resistance of the inlet system through which the air is admitted into the cylinders. The degree of filling a cylinder with the air varies with its temperature, engine speed and load, certain features of engine construction and service conditions.

In an actual engine, no adiabatic compression (one without a heat exchange with the surroundings) can possibly exist. It is rather characterised by heat exchange of varying intensity and direction between the gas and the surroundings. A change in the pressure and temperature of the compressed air charge and a decrease in the area of heat transfer, as the piston travels towards TDC, introduce much ambiguity into the realization of the mechanism of heat transfer. Actual compression follows a polytropic curve having a continuously variable exponent. Yet, it has much in common with the isothermic (constant temperature) and adiabatic processes due to the high rate of compression of the air charge.

The process of heat input, as this is assumed to occur in an ideal cycle, cannot be even approximately compared with the actual combustion taking place in the engine. It can be accounted for by a number of reasons. A point to be noted before all is that the combustion of fuel involves complicated physical and chemical changes. At its final stage, the temperature is at a maximum and so are the thermal losses.

As mentioned above, actual combustion overlaps to some extent the expansion stroke because of the rapidly increasing cylinder volume. This incurs extra losses of heat into the surroundings, impairing the effectiveness of heat utilization in the diesel engine.

The expansion of gases also follows a polytropic curve having a variable exponent but the mechanism of heat transfer, confusing as it is, becomes even more confusing at this stage owing to the fact that some of the fuel is continuing to burn during the expansion. The compressed and hot gases with unused energy escaping through the exhaust passages, which are set open before the piston arrives at BDC, contribute to additional heat losses.

An actual engine is bound to use some of its energy to drive the auxiliaries (lube oil, jacket water and scavenging pumps, etc.) and to overcome the friction between the rubbing components. Apart from that, it cannot operate without an effective cooling of the combustion chamber walls which is a further source of heat losses.

14. Engine Work and Power, Main Criteria of Engine Economy

It is known from physics that work is measured by the product of the force and distance. For determining the work of an engine in this way, it is necessary to know an arbitrary, but constant and mean value of the pressure in the cylinder, rather than the actual one which is changing at all times. Appropriate for the use in this case is a quantity termed the *mean indicated pressure* p_i. It is a pressure which, on acting upon the piston, performs the same work as the actual pressure in the course of the operating cycle.

The mean indicated pressure of the cycle can either be determined graphically from a diagram or be calculated from engine parameters. Knowing the theoretical mean indicated pressure, let us deduce a formula of power applicable to diesel engines of all types.

To solve this problem, construct a rectangle of an area equalling that of the indicator diagram. Referring to Fig. 6*b*, the height of the rectangle is the mean indicated pressure, p_i, and the length is the piston stroke.

The so-called indicated work is one performed in a cylinder of an engine per stroke

$$L_i = p_i \frac{\pi D^2}{4} S$$

where D = cylinder bore, cm; S = piston stroke, cm.

The indicated work per minute is

$$L_i = p_i \frac{\pi D^2}{4} SnK$$

where n = engine speed, rpm; K = stroke factor (for four-stroke engines, $K = 0.5$; for two-stroke engines, $K = 1$).

The indicated work per second, referred to as the indicated power N_i (in horse power) is given by the formula

$$N_i = \frac{p_i \pi D^2 S \cdot 10^4 nKz}{900}$$

where z = the number of cylinders.

Since the quantity $\frac{1}{60\times 75}$ is a constant, the indicated power of a four-stroke single-acting diesel engine can be determined by the

simplified formula

$$N_i = \frac{p_i V_s n z}{900}$$

where

$$V_s = \frac{\pi D^2}{4} S \text{ cm}^3$$

For a two-stroke single-acting diesel engine

$$N_i = \frac{p_i V_s n z}{450}$$

The effective power is determined by

$$N_e = N_i \eta_m$$

where η_m = mechanical efficiency of the diesel engine.

The existing nomenclature of effective powers, as stipulated for marine diesel engines by the relevant standard, is as follows:

rated power—a continuous effective power as specified by the manufacturer for rated crankshaft revolutions with a given set of auxiliaries under normal service conditions with provision for overload;

overload power—a short-time effective power in excess of the rated one with the same set of auxiliaries and under the same service conditions which can be used periodically for a limited interval only;

gross power—a continuous effective power guaranteed by the manufacturer for appropriate crankshaft revolutions with the same set of auxiliaries and under normal service conditions without allowance for overload;

minimum power—a lowermost effective power guaranteed by the manufacturer for appropriate crankshaft revolutions (corresponds to "SLOW" engine speed).

The rate of crankshaft rotation at a given irregularity factor is referred to as the minimum stable engine speed. A speed below this is likely to cause the stalling of the engine.

As already pointed out, some of the power developed by an engine goes to overcome the friction in the shaft bearings, engine cylinders, crossheads. Power is also wasted in order to impart motion to the lube oil, jacket water and scavenging pumps. The balance of power available at the output end of the engine, i.e. at the crankshaft flange, is termed *effective power*. It is lower than the indicated power by an amount corresponding to the mechanical losses as expressed in terms of mechanical efficiency which is the ratio of effective power and indicated one. The *mechanical efficiency* η_m is commonly between 0.75 and 0.85, varying with the number of strokes, the standards of

workmanship observed in building the engine, the number of engine-driven auxiliaries and the rate of boosting.

The degree of heat utilization is indicated by *thermal efficiency*, η_t, which is approximately 0.60 for an ideal diesel engine. The thermal efficiency of an actual engine is even less, being determined by the *indicated efficiency* η_i.

The ratio of the heat converted into actual work and the total heat supplied is termed *actual efficiency* η_a. Allowing for all thermal and mechanical losses, the actual efficiency indicates how efficiently heat is utilized in the engine

$$\eta_a = \eta_i \eta_m$$

Accordingly, the mean effective pressure in an engine cylinder is

$$p_e = p_i \eta_m$$

The actual efficiency is a yardstick of engine economy. Being 0.35-0.45 for the four-strokes and 0.32-0.42 for the two-strokes, it indicates that the former are more economical than the latter.

Engine economy can also be expressed in terms of *specific fuel consumption*. It can be calculated if we know the indicated or actual efficiency and the net calorific value of the fuel, Q_n.

Taking that the indicated specific fuel consumption is

$$g_i = \frac{860}{\eta_i Q_n} \text{ g/kW h}$$

or

$$g_i = \frac{632.3}{\eta_i Q_n} \text{ g/hp h}$$

the actual specific fuel consumption may be computed by the formula

$$g_a = \frac{g_i}{\eta_m} \text{ g/kW h (g/hp h)}$$

Practical values of the actual specific fuel consumption are as follows:

four-stroke diesel engines, 224-232 g/kW h (145-170 g/hp h);
two-stroke diesel engines, 225-252 g/kW h (155-185 g/hp h);
carburetor engines, 306-340 g/kW h (225-250 g/hp h).

15. Heat Balance of a Diesel Engine

The concept of heat balance is introduced to show how the heat liberated in an engine due to the combustion of fuel is utilized. Taken as 100%, it is the equivalent of useful work and various losses.

Heat balance is essential for studying the thermal processes in the engine, increasing the efficiency of its cooling system or determining the heat transfer area of a waste-heat equipment in the exhaust duct. The elements of the heat balance of a marine diesel engine are illustrated in Fig. 14. They are expressed either in units of heat or percent of the heat supplied.

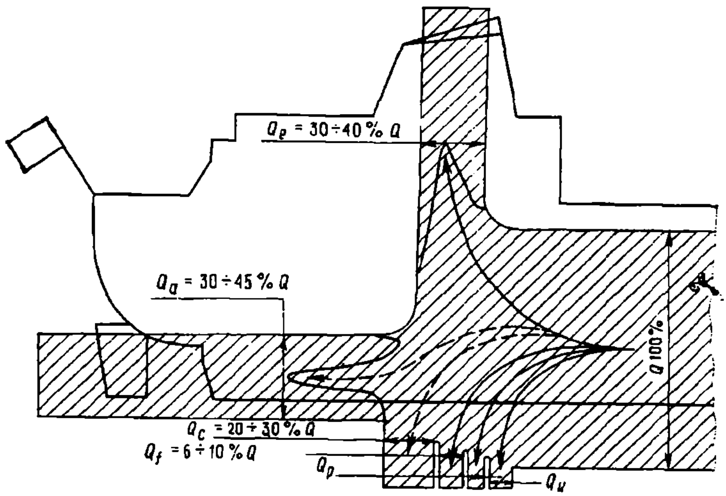


Fig. 14. Heat balance of marine diesel engine

The data representing heat balance are acquired experimentally from testing the engine under stable thermal conditions. The main components of the heat expended are as follows:

—the heat Q_a, converted into actual work (accounts for 30-45% of the total heat supplied due to the combustion of fuel);

—the heat Q_e, lost with the exhaust gases;

—the heat Q_c, lost through the cylinder walls due to cooling;

—the heat Q_f, lost due to the friction of parts;

—the heat Q_p, lost due to the partial combustion of fuel;

—the heat Q_u, lost due to other losses unaccounted for.

Thus, the heat balance can be written as

$$Q = Q_a + Q_e + Q_c + Q_f + Q_p + Q_u$$

The way in which the heat supplied is distributed between the components of the heat balance depends on such factors as engine speed and size, cooling method, the use of supercharging, etc. So, in a supercharged engine, the total heat Q supplied to the cylinders increases and the losses with the coolant decrease by 10-12%, while

the amount of heat lost with the exhaust gases rises to about 40-45%.

The thermal stresses set up in the cylinder walls and pistons vary directly with the mean piston speed and inversely with the ratio of piston stroke to cylinder bore.

To ensure normal engine performance, an effective cooling system is indispensable, implying that the losses of heat with the coolant are unavoidable.

To recover the heat of the exhaust gases leaving two-stroke diesel engines at 523-573 K (250-300°C) and four-stroke ones at 673-723 K (400-450°C), use is made of waste-heat steam generators which produce hot water or steam at a pressure of 49×10^4 to 78.5×10^4 N/m^2 (5-8 kgf/cm^2). This significantly increases the amount of heat converted into useful work. The waste-heat steam generator is commonly installed in the funnel, serving as a double of a spark arrester and a muffler.

REVIEW QUESTIONS

1. What is a four-stroke engine?
2. What processes take place in a cylinder of a four-stroke diesel engine with each stroke?
3. What is a two-stroke engine?
4. What processes take place in a cylinder of a two-stroke diesel engine?
5. What is the operating principle of a carburetor engine?
6. Contrast the advantages and disadvantages of two- and four-stroke engines.
7. What is the purpose of advancing the opening of the exhaust valve and delaying the closing of the inlet valve in a four-stroke engine?
8. Why is the fuel injected into the working cylinder of a diesel engine with some advance?
9. What is valve overlapping?
10. What phenomena take place in a cylinder of a diesel engine during the period of ignition lag?
11. What cycles are referred to as ideal and what purpose do they serve?
12. Compare the ideal cycles of diesel engines in terms of economy.
13. What is the difference between ideal and actual cycles of engines?
14. What is mean indicated pressure?
15. What are indicated mechanical and actual efficiencies?
16. How can one determine the indicated and actual power of an engine?
17. What purpose serves the heat balance of an engine?

Chapter III

INTERMIXTURE AND ATOMIZATION OF AIR AND FUEL CHARGES IN DIESEL ENGINES

16. Air-Fuel Intermixture in Airless-Injection Diesels

Modern diesel engines are all of the airless-injection type which can afford a very brief period for the combustion to take place. For example, in high-speed diesels combustion lasts only 0.003 s, maximum 0.015 s. Thus, to ensure complete combustion within the time available is a difficult task.

To obtain complete combustion, the fuel charge is to be atomized into a mist through the small orifices of the injector, mixed with air, heated up and vapourized, partly or completely. The high pressure required for atomization is provided by the fuel pump of the engine.

The air-fuel intermixture is accomplished either volumetrically or by way of a film. In the former case, the injected fuel intermixes with the air directly in the combustion chamber before vapourizing. In the latter case, the injected fuel vapourizes on contact with the hot piston crown (its temperature is 620-720 K) spreading as a thin film, and the vapour so formed mixes with the air by virtue of a swirl set up inside the combustion chamber. The rest of fuel which has failed to vapourize intermixes with the air and vapourizes volumetrically. The intermixture of the air and fuel charges by way of a film is commonly employed in engines with the cylinder bore under 180 mm.

Whatever the method of air-fuel intermixture, the fuel is being injected into the cylinder at a velocity of 150-500 m/s, forming a cone-shaped spray whose length varies directly with the injection pressure. This pressure is $1\,180 \times 10^4$-$4\,900 \times 10^4$ N/m^2 (120-500 kgf/cm^2) on low- and medium-speed diesels and $5\,880 \times 10^4$-$14\,700 \times 10^4$ N/m^2 (600-1 500 kgf/cm^2) on high-speed ones.

To obtain normal injection, it is necessary that the length of the fuel spray is correlated with the depth of the combustion chamber so that the fuel spray reach neither the piston crown nor the cylinder walls. This will prevent incomplete combustion and carboning which lead to excessive fuel consumption and overheating of the piston. On the other hand, a short fuel spray is also a drawback, for the air will not mix up with the fuel, failing thus to take part in combustion, and the engine will be incapable of attaining the rated power.

Theoretically, 1 kg of fuel requires 14.5 kg of air to burn up completely. However, the practical air requirement is higher, for other-

wise it would be impossible to run the engine economically under poor air-fuel intermixture or overload. To take this condition into account the so-called *excess-air coefficient*, α, is introduced which is the ratio of the actual amount of air and the theoretical one required to burn 1 kg of fuel. On diesel engines, α varies within 1.3 and 2.2 to achieve complete combustion.

17. Methods of Stirring Up Air-Fuel Intermixture

Shipshape order of the fuel system cannot alone provide for an orderly mixing of the air and fuel charges. The thoroughness of air-fuel intermixture also depends on the shape of the combustion chamber, scavenging method, engine speed and some other factors. The air-fuel intermixture can be stirred up by shortening the intervals required by the droplets of fuel to heat up and vapourize in the combustion chamber. Helpful in this case are refinements of the design, some aspects of physical and chemical nature, etc. By way of example, a doped fuel of a carefully chosen chemical composition can improve combustion. To shorten the ignition lag, steps are to be taken so as to increase the initial pressure and temperature, the compression pressure and temperature, the compression ratio and the excess-air coefficient. The same effect may be obtained by applying a heat-resistant coating to the piston crowns, by fitting them with inserts, or by atomizing the fuel in a more finer way.

Obviously, any step taken to streamline the process of air-fuel intermixture and speed up combustion is conducive to a higher efficiency of the combustion. Frequently resorted to in order to promote the intermixing of the air and fuel charges is swirling motion of the incoming air in the engine cylinder. Apart from turbulence (swirl) chambers of various kinds with air storage chambers and hot surfaces, also in use on four-strokes are masked valves.

Research and development work currently in progress is aimed at intensifying the process of the intermixing of fuel and air charges with the aid of ultrasound and ionization relying for operation on the battery devices. They are designed to ensure complete combustion of the fuel, boost the actual efficiency of diesel engines and minimize the deposition of carbon on the walls of combustion chambers.

18. Modes of Air-Fuel Intermixture

The intermixing of the air and fuel charges may be accomplished in a variety of ways. Therefore, distinction is made between

—direct-injection intermixture;
—precombustion-chamber intermixture;
—turbulence (swirl)-chamber intermixture;

—auxiliary-chamber intermixture.

Let us consider the mechanism of intermixture observed in each particular case.

The so-called direct-injection intermixture depicted in Fig. 15*a*, *c* *d* and *e* is in use on medium- and high-output diesel engines employing piston crowns of various configuration. The most simple one to make is the flat-head type. However, the combustion chamber is

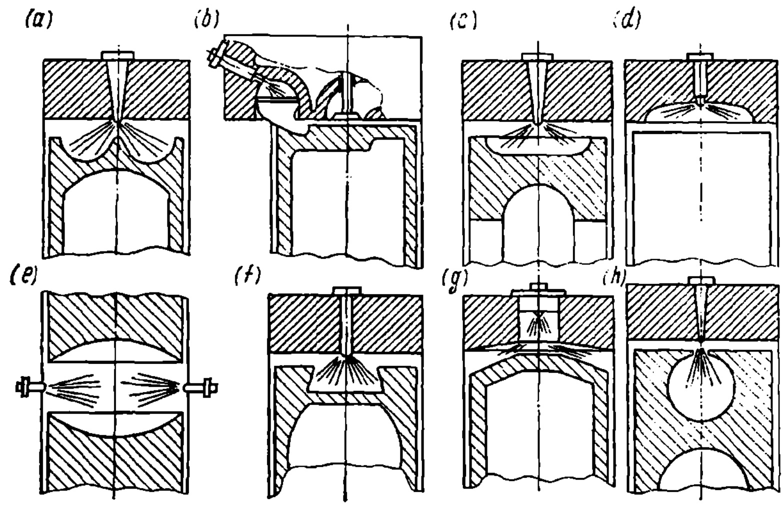


Fig. 15. Typical configurations of combustion chambers

of a shape which correlates with neither the direction nor penetration of the fuel spray, giving rise to significant heat losses. This condition should only be met when the fuel particles burn up in the air before reaching the walls of the combustion chamber. On the engines employing the direct-injection intermixture perfect fuel atomization takes place due to the injection pressure being as high as 196×10^5-980×10^5 N/m^2 (200-1 000 kgf/cm^2) and even higher. Three to ten fuel injectors are commonly used with the orifices varying in diameter between 0.15 and 0.5 mm to ensure a uniform distribution of the fuel spray through the combustion chamber.

The pistons of high-output diesel engines have dome-shaped heads. Their upward extending rims safeguard the upper belt of cylinders against the impinging fuel spray, preventing the carboning up of the liners (Fig. 15*a*). The combustion chamber confined by the dome-shaped piston head promotes a swirling air motion in the cylinder during the compression stroke.

To prevent incomplete fuel combustion resulting from non-uniform distribution of the fuel charge in an open combustion chamber, the diesel engine with the direct-injection intermixture commonly operates at an increased excess-air coefficient, $\alpha = 1.7$-2.2. By virtue of the small area of heat transfer in an open combustion chamber keeping the heat losses low, the direct-injection engines are economical and can be started with ease. The disadvantages are rough and noisy running and the need in high-performance fuel equipment.

The precombustion-chamber mode of the intermixture of the air and fuel charges (Fig. 15*f*, *g*, *h*) is in use on diesel engines with a cylinder bore of 180-200 mm. It is accomplished with the aid of a separate pre-chamber, or pre-combustion chamber, located in the cylinder head and communicating with the main chamber through small passages of 4-6 mm in diam which vary in number between one and twelve. The volume of the pre-chamber is commonly 25-40% that of the main chamber so that there burns from 10 to 15% of the fuel charge. The balance of the fuel unable to burn due to the lack of air is expelled through the passages into the main chamber together with the products of combustion. The high velocity of the flow entering the main chamber facilitates the intermixture of the unburnt fuel and gases with the air in the cylinder, promoting complete combustion. Since the combustion in the main chamber is timed to occur unsynchronously with that in the pre-chamber, the pressure builds up in the cylinder gradually, relieving the engine of rough running. The maximum combustion pressure is 490×10^4 to 590×10^4 N/m^2 (50-60 kgf/cm^2).

The precombustion-chamber intermixture requires no high-pressure multi-orifice fuel injectors. Typical for this mode of intermixture are injectors with the orifice of from 0.5 to 1.0 mm in diameter which inject fuel under a pressure of 980×10^4-$1\,470 \times \times 10^4$ N/m^2 (100-150 kgf/cm^2). The engines relying on the precombustion-chamber intermixture are less demanding in high-grade fuels and run at an excess-air coefficient lower than is the case with the direct-injection engines ($\alpha = 1.2$-1.6).

Along with advantages, the precombustion-chamber intermixture exhibits the following drawbacks. Before all, the starting is difficult. The point is that the compression temperature commonly tends to decrease due to the losses of heat through the combined surface area of the main chamber and the pre-chamber which is significant. For a failure-free starting in use are various starting aids, e.g. electric heater plugs, flame primers, etc. The precombustion-chamber engines are less economical than the direct-injection units, apparently because of the losses of energy in the connecting passages.

The turbulence (swirl)-chamber intermixture of the air and fuel charges (Fig. 15*b*) is taken care of also by a separate chamber of

spherical or cylindrical shape termed turbulence (swirl) chamber. It is connected to the main chamber through a single passage which extends tangentially with respect to the walls of the swirl chamber and is larger in cross section than the passages of the precombustion chamber. The peculiar shape of the turbulence chamber and the tangential arrangement of the interconnecting passage are the factors inducing the incoming air to follow a helical path. The bottom of the turbulence chamber is made of a heat-resistant steel and functions as a hot surface. Heating up during the combustion stroke, this rejects its heat to the incoming air. The injectors used in this case are of the single-orifice or pintle-type with an orifice diameter of 0.6-3 mm which inject fuel under a pressure of 980×10^4-$1\,370 \times 10^4$ N/m^2 (100-140 kgf/cm^2) into the swirl chamber. The fuel injected evaporates and ignites. However, a lack of air prevents complete combustion, for the volume of the swirl chamber accounts only for 40-50% of that of the main chamber. The high-pressure gases formed expell the unburnt fuel into the main chamber where it mixes up with air. In this way complete combustion is obtained without a too high excess-air coefficient.

The disadvantages of the turbulence chamber intermixture are the same as in the precombustion-chamber mode: high fuel consumption and difficult starting. They arise because of the losses of heat through the walls of the turbulence chamber and are coped with by the recourse to the starting aids referred to above.

A piston crown of peculiar shape, the combustion chamber being located almost totally therein (Fig. 15*h*), is a salient feature of the auxiliary-chamber mode of intermixture. A passage of a diameter from 1/3 to 1/2 that of the cylinder bore, interconnects the combustion chamber and the cylinder space. The high-velocity stream of hot air and gases emerging from the auxiliary combustion chamber thoroughly intermixes with the fuel in the cylinder, initiating ignition and promoting complete combustion. Owing to this arrangement, it is possible to burn 20-25% more fuel in the same cylinder volume and obtain similar increase in the engine's power output.

In conclusion, it should be pointed out that the direct-injection mode of air-fuel intermixture combined with the open combustion chamber requires a thorough filtration of the fuel. The fuel equipment used to produce the effect of intermixture with the aid of precombustion, swirl and auxiliary chambers is less sensitive to the fuel filtration quality and is therefore more simple, dependable and durable than fuel equipment of direct-injection diesel engines.

REVIEW QUESTIONS

1. What is ignition lag?
2. How is the intermixture of the air and fuel charges effected?
3. Why is an ignition advance necessary?

4. What is the difference between the volumetric air-fuel intermixture and the intermixture by way of a film?
5. What is excess-air coefficient?
6. Discuss the configurations of the combustion chambers used on diesel engines.
7. Discuss the direct-injection, precombustion-chamber, turbulence (swirl)-chamber and auxiliary-chamber modes of intermixing the air and fuel charges. Contrast their advantages and disadvantages.

Chapter IV

FUELS AND LUBRICATING OILS FOR DIESEL ENGINES

19. Petroleum and Products of Its Distillation

Crude petroleum is a thick oily liquid, yellowish-green to black in colour. It is a complex mixture of liquid hydrocarbons with organic compounds containing oxygen, nitrogen and sulphur.

The bulk of petroleum produced consists of hydrocarbons occurring in three principal types: aromatic hydrocarbons, naphthenes and saturated paraffins.

Aromatic hydrocarbons are chemically-instable liquids with a ring-type molecular structure. The hydrocarbons of the paraffin series are chain compounds occurring in gaseous, liquid and solid state. Paraffins are saturated hydrocarbons which are characterized by a high chemical stability at normal temperatures but are apt to oxidize at 473-523 K (200-250°C). The hydrocarbons of the naphthene series are also chain compounds less stable, however, than paraffins.

Petroleum, after a complex treatment at the refinery, yields an extensive range of liquid fuels and lubricants. What is called straight-run distillation in a vacuum is a treatment intended to produce low-boiling fractions free from unwanted by-products. Each of the fractions is separated at a given temperature. So, for example, the temperature range between the boiling point and 473-498 K (200-250°C) yields the gasoline fractions, the naphtha fractions are separated at 393-503 K (120-230°C), the kerosene fractions are distilled at 423-588 K (150-315°C), and the gas oil fractions boil at 513-623 K (240-350°C).

The range of petroleum products obtained by straight-run distillation at the appropriate temperature is as follows:

—petroleum ether, 313-368 K (40-95°C);
—aviation gasoline, 313-453 K (40-180°C);
—motor gasoline, 313-473 K (40-200°C);
—naphtha, 393-513 K (120-240°C);

—turbine fuel, 423-588 K (150-315°C);
—diesel fuel, 463-623 K (190-350°C);
—gas oil, 503-633 K (230-360°C);
—burner fuel, 573-673 K (300-400°C).

The residual of distillation is either converted to gasoline and other low-boiling fractions by thermal cracking or is used as boiler fuel.

The products of distillation are treated with alkalies (sodium hydroxide, etc.), sulphuric acid or bleaching earths to remove nitrogen and sulphuric compounds, oxides and other harmful admixtures.

20. Properties of Diesel Fuel

Annual fuel consumption rises steadily by 5-6%. In the light of this fact, the substitution of cheap viscous fuel for the difficult-to-obtain diesel fuel helps solve the problem of cutting operating costs which has recently become the source of main concern among diesel engine builders. A thrifty use of the right grade fuel selected with due care may reduce the operating costs to 15-20% of the overall expenses for the ship's upkeep.

The fuel used in marine diesel propulsion plants must meet the following requirements:

—good combustibility;
—freedom from an explosive hazard;
—combustion without the carboning up and corrosion of parts;
—fluidity at low temperatures ensuring trouble-free fuel supply to the pump and injectors.

Marine diesel engines are run on either pure distillate fuels or their blends. Since the performance of a diesel is directly influenced by fuel, suitable for use are only those grades which meet the specifications outlined in relevant standards.

Consider the main properties of fuel referred to in the specifications.

The fractional composition should be specified for all diesel fuels and gasolines, with volatility given for each fraction at different temperatures. Fractional composition influences the starting of engines from cold, fuel combustion and consumption, and the carbon-forming tendency. Low-boiling fractions reduce the flash point which, this should be borne in mind, must not be less than 338 K (65°C) according to the existing rules. An actual distillation temperature which is less than the specified one indicates that the low-boiling fractions exceed an allowable minimum. The fuel with a low distillation temperature must be rejected, for—apart from the above reason—it causes the carboning up of injectors and smoky exhaust. On the other hand, an excessive content of high-boiling fractions impairs the air-fuel intermixture, inviting incomplete combustion. But heavy

fuels are more safe in point of explosiveness due to their poor volatility.

Viscosity defines the resistance due to the friction between the layers of the working fluid, which move relative to each other under the influence of an applied force. Viscosity is a very important property affecting the fuel's atomizing, filtering and pumping capacity. The higher the viscosity the more reluctant is the fuel to filter and atomize. However, viscosity can be reduced by heating up the fuel.

Distinction is made between three types of viscosity: specific, kinematic and dynamic.

Specific viscosity is the ratio of the efflux time of 200 cm^3 of fuel at 293-323 K (20-50°C) and that of 200 cm^3 of distilled water at 293 K (20°C) as determined by means of a viscometer with a 2.8-mm orifice. The unit is the degree of specific viscosity.

Dynamic viscosity is the viscosity of a substance in a laminar, i.e. streamlined flow containing layers spaced one centimetre apart, which require a tangential force of one dyne per square centimetre to be moved at velocities differing by one centimetre per second. The unit of dynamic viscosity is the poise, P; 1 P = 0.1 Pa s.

Kinematic viscosity is the ratio of dynamic viscosity and the density of the fluid at the same temperature. The unit is the stokes, St, but for convenience the centistoke, cSt, equal to 1/100 stoke is used. In the USA, the unit of kinematic viscosity is the Saybolt seconds Universal and in Great Britain, the Redwood seconds No. 1 and No. 2. However, there is a tendency to use the centistoke.

Carbon residue indicates the tendency of a fuel to form carbon deposits. The unit sometimes in use is the coke value which should not exceed 0.05-0.1%. A high carbon residue is a cause of piston and cylinder wear. It also brings about the plugging of injectors. To avoid this, fuel is doped with additives preventing the deposition of unsoluble particles of carbon therefrom.

Sulphur content is chief consideration in evaluating the corrosion-inhibiting properties of fuel and its quality as a whole. Sulphur in the fuel is harmful, for the sulphur dioxide resulting from the fuel's burning combines with water vapours to form sulphuric acid which attacks the cylinder liners, piston rings and fuel equipment, causing their corrosion. Sulphur also forms a hard deposit on parts, whose removal is a problem. Residual fuels with a sulphur content of 3.5-4% can be used on low-speed diesels only, provided the crankcase oil is of a strongly-alkaline grade producing a neutralizing effect.

The flash point is the minimum temperature at which the fuel is capable of volatilizing and the air-fuel mixture is capable of igniting on contact with an open flame. The flash point is the main criterion for the fire-hazard classification of fuels. Low-boiling fractions depress the flash point.

The self-ignition point is the minimum temperature at which the fuel is capable of igniting on its own accord without the agency of a heated body or flame. This temperature must be taken into consideration in choosing the compression ratio of an engine and an appropriate fuel grade for the engine.

The pour point is the temperature at which the fuel ceases to flow. The importance of this property needs no emphasis in contemplating the requirements to be met by the facilities and equipment for storing, heating up and pumping the fuel under the conditions of low temperatures in order to avoid the crystallization of the wax contained therein. Wax crystallization renders pumping difficult, if possible at all.

The calorific value is the amount of heat produced by complete combustion of one kilogram of fuel. The calorific value of viscous fuels is low. This is exemplified by the fact that one kilogram of diesel fuel liberates 10 100-10 300 kcal whereas the same amount of marine residual fuel generates 9 500-10 000 kcal, i.e. by 350-400 kcal less (10 000 kcal = 41 900 kJ/kg).

The cetane number is an index of ignition quality of the diesel fuel which defines the way in which combustion proceeds in the cylinders. It is determined by comparing the ignition quality of a standard solution, which is a mixture of two hydrocarbons termed cetane and alpha-methyl naphthalene, with the ignition quality of the fuel tested. The cetane number is the amount of cetane in per cent contained in the standard solution which has an ignition delay equalling the ignition delay of the fuel tested. The higher the cetane number, the shorter the ignition delay and the easier the starting of the diesel. The cetane number of the diesel fuels in use is commonly 35-55.

21. Fuels Used by Marine Diesel Engines

Marine diesel engines are run on low-viscous diesel fuels and on heavy residual fuels. Each engine must be assigned to use a proper grade of fuel depending on its application and speed with provision for minimum injector carbonization and easy starting.

Medium- and high-speed diesels commonly use diesel fuels with a low or medium viscosity. Low-speed diesels may employ either diesel fuels or residual fuels, except for those with the crankshaft rate of rotation between 100 and 350 rpm which are run on residual fuel only.

Residual fuels, if used, call for effective separation of the sediment and heating up of the fuel. A careful tuning up of the fuel system and the use of lubricating oils with additives neutralizing the corrosive effect of sulphur are also very essential in this case.

The fuel, which has been settled and centrifuged, is heated by

coils fitted in the fuel tanks close to their bottoms. The permissible temperature of heating must be at least 10°C less than the flash point of the fuel vapour, and the pressure of the steam or hot water in the heating coils must be not over 68.5×10^4 N/m^2 (7 kgf/cm^2). The fuel filters and fuel lines must be provided with steam jackets to ensure fluidity of the fuel.

The quality of the fuel used is the main factor influencing the wear on the fuel equipment, cylinder liners, pistons and piston rings. It also has a decisive effect on the length of periods between overhauls, maintenance costs and the cost of spare parts.

In this country, marine diesel engines are run on automotive diesel fuels of the following grades:

—arctic fuel A for the ambient temperatures of 223 K (—50°C) and upwards;

—winter fuel З (W) for the temperatures of 243 K (—30°C) and upwards;

—summer fuel Л (S) for the temperature of 273 K (0°C) and upwards;

—special fuel C (X) (used in indoor-operating engines).

Also in use in this country are diesel fuels of straight-run distillation, differing from the automotive grades by a higher flash point and lower sulphur content which is 0.2% maximum. They are classed as follows:

—arctic fuel ДА for use at the temperature below 243 K (—30°C);

—winter fuel ДЗ (W) intended for use at the ambient air temperatures over 243 K (—30°C);

—summer fuel ДЛ (S) for the temperatures over 273 K (0°C);

—special fuel ДС (X) for indoor operating diesel engines.

22. Bunkering of Fuels and Lubricants

The bunkering of fuel and lubricants is an operation requiring careful attention, for trouble-free operation of the ship's propulsion plant depends to a considerable extent on its success. The bunkering may be accomplished from a shore facility or from a fuel barge.

Before each bunkering or every six months, whichever is earlier, the ship's fuel tanks need washing and cleaning. On sea-going ships this interval may be two years.

Prior to bunkering, check the bunkering and fire-fighting equipment for condition, paying particular attention to the hoses, seals, filling holes, strainers. Make sure the hose unions tightly fit the filling holes and the fuel tanks contain no water. Only battery-powered lamps of the explosion-proof type should be used inside the tanks for illumination. The personnel must be provided with means preventing their poisoning with the gases.

In the course of bunkering, keep all the fire-fighting equipment on standby (applies to foam extinguishers, sand containers, etc.). Smoking should be strictly forbidden, and a "Fire Hazard" caution sign must be placed next the filling hole on deck.

The bunkering of fuel and lubricants is chief engineer's responsibility; he also fills out the forms. Directly in charge of the operation is either the third engineer or the engineer of the watch. The officer in charge signals the beginning of bunkering or a change in the flow rate of fuel and monitors the progress of the operation by gauging the ship's tanks and the tanks of the bunkering facility.

Accepted for bunkering must be the certified fuel only. The fuel certificate is to be checked against an appropriate standard or the engine service manual to be sure that the fuel is of the specified grade.

23. Fuel Consumption Rates and Fuel-Saving Practice

The rated fuel consumption of each particular engine is indicated in the manufacturer's instructions. The actual fuel consumption betrays the actual state of the engine. If it is too high, the reason may be engine overload, poor repair of the fuel equipment, excessive power consumption by the auxiliaries or high exhaust back pressure due to the carboning up of the ducts.

Any diesel engine must be run strictly in accordance with the instructions outlined in the service manual. This is a prerequisite that the fuel consumption should be within the rated value.

To keep the fuel consumption at the rated value and effect economy of fuel, it is good practice to check the fuel equipment for condition and pressure-test the injectors at regular intervals.

However, the fuel consumption may increase with an increase in the resistance to ship motion which may be caused by a too deep draught, heavy head wind, bad fouling of the ship's hull and strong return current. An increase in the fuel delivery per cylinder, in order to overcome the high ship resistance at full speed, causes a rise in the indicated mean pressure with the result that the engine is overloaded. To avoid this, for diesel engines are very susceptible to overloads, it is essential to set the fuel pump for normal fuel delivery and to avoid accelerating the engine's rotation to full speed. Thus, in the case of an increased ship resistance to motion the amount of fuel delivered per cycle should be reduced so that the diesel performance is confined to the specified limits.

In a direct-coupled engine, an increase in the engine speed may lead to an overload even at normal indicated mean pressure, or inversely, an increase in the indicated mean pressure may result in an overload while the engine speed is normal. In either case there is an increase in the specific fuel consumption, rapid wear of the

rubbing components, thermal overstressing of the cylinders and valves, poor lubrication, incomplete combustion with an overlapping of the expansion stroke by the combustion stroke, which result in smoky exhaust and intensive carboning up of the cylinders. These drawbacks resulting from overloads are unavoidable and manifest to a varying degree in all internal-combustion engines.

High-speed diesel engines designed to operate at low excess-air coefficient and a higher than normal indicated mean pressure are capable of sustaining 1-hour 10% overloads.

The generally accepted sign of overload is smoky exhaust. However, this is true only of the engine in good repair. Not infrequent are the cases, where the cylinders share unequally the engine load because of natural or untimely wear of the fuel equipment and the engine's various moving components. Some of the cylinders become thus overloaded and produce a smoky exhaust although the engine as a whole may be underloaded. In a situation like that one, smoky exhaust will betray poor repair of some engine components.

An inconsistency between the characteristics of a constant pitch propeller and the rated engine power may also give rise to an overload with the result that the engine will fail to develop the crankshaft revolutions corresponding to a normal fuel delivery as obtained from the tested trials of the injection pumps by the manufacturer.

A similar picture, an evidence of overload, may be observed when heavy deposits of ash, sludge and carbon in the exhaust duct impede the outflow of the burnt gases. A restricted cross section, sharp bends and intricate layout of the exhaust duct or an undersized muffler are other causes of engine overload.

Not to be overlooked is the fact that the power and economy of marine engines vary with the temperature of incoming air and barometric pressure, other things being equal. When air is being drawn through the deck downcasts, its temperature may vary in the range from 253 K (—20°C) to 318 K (45°C) and the pressure from 720 to 770 mm Hg depending on the navigation area. As a result, the mass of the air reaching the cylinders may decrease by 12-15%, reducing the excess-air coefficient and impairing engine power and economy.

24. Requirements for the Use of Residual Fuel

As mentioned above, the tendency in marine practice to cut operating costs has lead to a large-scale application of cost-saving viscous fuels priced at one half or even less of the diesel fuel.

However, residual fuel, if not conditioned, lends itself difficult to atomization and burns incompletely. This gives rise to extra thermal stresses, carbons up the cylinders and pistons and leads to their rapid wear. The high viscosity of residual fuels poses problems

during their pumping and cleaning, calls for using high injection pressures.

With reference to their viscosity residual fuels fall into two groups: medium- and high-viscosity fuels. Their viscosity at 50°C is 36.0 and 150 cSt, respectively.

Medium-viscosity fuels include motor fuel DT, marine mazoutes F-5 and F-12, while high-viscosity working fluids, motor fuel DM, mazoute "+10", and burning mazoute M-40 (ref. Table 1). Unlike diesel fuels, residual fuels are characterized by high contents of sulphur, water, sediments, asphaltenes. They also contain certain amount of vanadium and nickel.

A diesel intended to be run on residual fuel must be provided with means of lessening its viscosity to an acceptable level. The heating up of the fuel in service and settling tanks is a difficult task because the fuel temperature to be obtained depends on the initial viscosity of the fuel. The maximum temperature of fuel must be 15°C below the flash point but not over 363 K (90°C) or otherwise volatiles may form, creating an explosive hazard. The fuel heating used to stabilize viscosity may be automated, with the aid of controllers which admit hot steam into the heating coils in amounts decided by the actual viscosity of the fuel. Normally, its specific viscosity before the injectors is maintained at or under 4 deg, i.e. the optimum value is 1.7-2.5 deg. To keep the fuel fed to the injectors at the above viscosity, the lines are provided with thermal insulation or reheaters. The line pressure must be maintained in the range from 39.4×10^4 to 49.0×10^4 N/m^2 (3-5 kgf/cm^2) to prevent the vapourization of low-boiling fractions.

The sediments commonly present in residual fuel in an amount of 0.4-0.5% interfere with the normal functioning of the fuel equipment. A solid particle between the nozzle valve and nozzle body of an injector or the delivery valve and valve seat of the fuel pump may upset the injection. It is also conducive to rapid wear of matching components. To avoid this, sediments must be removed from residual fuel by way of settling, heating, filtration and two-stage centrifugation.

If ash would result from combustion, the compounds present in it form adherent deposits of oxides of iron and silicon which cause errosion of injector orifices, piston crowns and cylinder liners. Particularly harmful are the compounds of vanadium and sodium, causing the corrosion of supercharger blades and exhaust valves. Moreover, melting at 873 K (600°C), which is the temperature of the exhaust turbine blades and exhaust valves, they form a brittle coating on these parts. Particles of this coating, coming between an exhaust valve and its seat, bring about blow-bys and the scorching of the valve; they also interfere with the functioning of the exhaust turbine because of polluting its blades.

Table 1

Specifications for Diesel Fuels Used in the USSR

Test	Diesel fuel grades											
	ДА	ДЗ(W)	ДЛ(S)	ДС(X)	А	З(W)	Л(S)	С(X)	ДТ	ДМ	ТЗ(W)	ТЛ(S)
Cetane number, min	40	40	45	50	45	45	45	50	—	—	45	45
Distillation temp, °C, min for fractional composition:												
10%	200	200	—	—	—	—	—	—	—	—	—	—
50%	255	275	290	280	240	250	280	280	—	—	275	290
90%	300	335	350	—	—	—	—	—	—	—	—	—
96%	330	—	—	340	—	—	—	—	—	—	—	—
98%	—	—	—	—	330	340	360	340	—	—	340	360
Kinematic viscosity, cSt												
at 20°C	2.5-4.0	3.5-6.0	3.5-8.0	—	1.5 min	1.8-3.2	2.8-6.0	4.5-8.0	—	—	2.2-5.0	3.5-6.5
at 50°C	—	—	—	2.5-4.0	—	—	—	—	36.0	150	—	—
Specific viscosity, deg	—	—	—	—	—	—	—	—	5.0	20	—	—
Acidity, mg KOH/100 cm³ of fuel, max	5	5	5	5	5	5	5	5	—	—	5	5
Ash, %, max	0.01	0.02	0.02	0.02	0.01	0.01	0.01	0.01	0.04	0.15	—	0.01
Sulphur, %, max												
low-sulphur grade	0.2	0.2	0.2	0.2	0.4	0.6	1.0	1.0	0.5	0.5	0.5	0.5
high-sulphur grade	—	—	—	—	—	—	—	—	1.5	3.0	—	—
Sediments, %, max	—	—	—	—	—	—	—	—	0.1	0.2	—	—
Flash point in closed cup tester, °C, min	35	50	60	90	30	35	40	90	65	85	40	65
Pour point, °C, max	—60	—45	—10	—15	—55	—35	—10	—15	—5	10	—35	—10
Water, %, max	—	—	—	—	—	—	—	—	1.0	1.5	—	—

The sulphur present in fuel forms organic and inorganic acids during combustion which cause galvanic corrosion and rapid wear of the cylinder liners, pistons and piston rings. Since the removal of sulphur from fuel is a difficult problem, the maximum permissible sulphur of a residual fuel is 2.5-3.0%.

To combat sulphur-induced corrosion, steps are taken to prevent the condensation of the acids or neutralize them. The remedy is

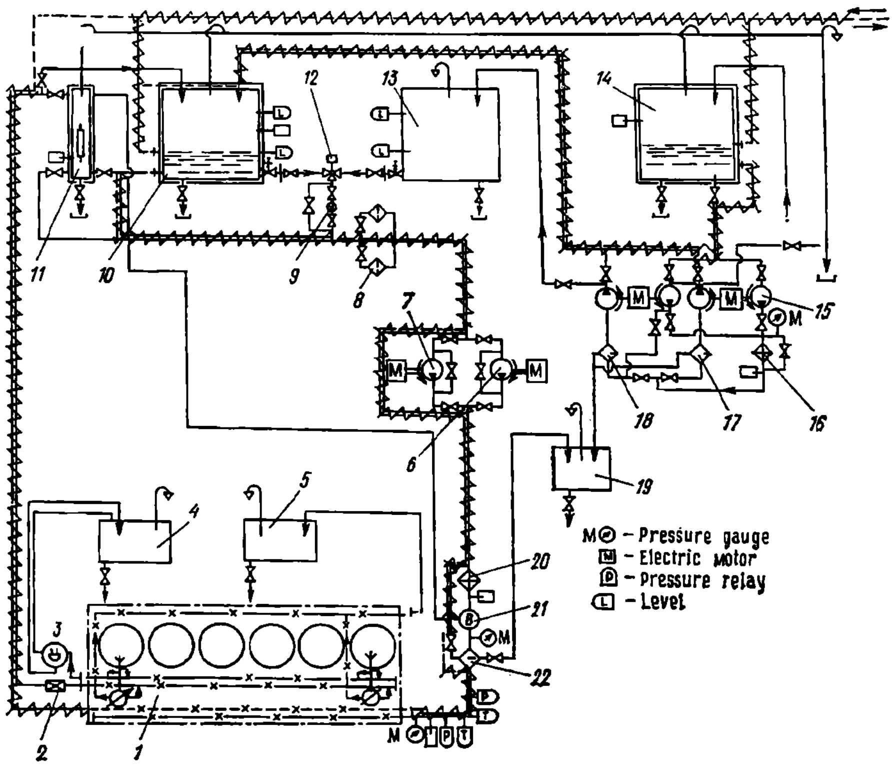


Fig. 16. Schematic of fuel system adapted for residual fuel

Key to pipelines: —— external fuel lines; — × — fuel lines on engine; — — × — — steam lines; thermally-insulated fuel lines

either to run the engine at a maximum cooling water temperature that permits adequate lubrication, or to use high-alkalinity crankcase oil.

Asphaltenes, if present in fuel, are a source of heavy carbonization of the injectors, exhaust ports and pistons. The carbon causes a rapid abrasive effect on the cylinders and pistons. To minimize this effect, it is good practice to make the cylinder liners from a wear-resistant

cast iron, giving their bores a coat of porous chromium, and to use high-quality crankcase oil.

If adapted to run on residual fuel the diesel engines should use diesel fuels when they are started, warmed up, maneuvered and shut down. Although this practice complicates the fuel system, for the tanks, fuel-purification equipment and fuel lines must be provided both for the residual fuel and for the diesel fuel, the economy gained more than offsets the installation and operation costs of such a system.

Residual fuel is used not only on low-speed diesel engines such as MAN KBZ 70/120, Fiat C758S, Sulzer Bros 5SAD-72, etc. Following suit are also some of the medium-speed diesels. This is exemplified by Pielstick diesels burning fuel oil with a viscosity of 3 500 seconds Redwood at 100°F, which is the equivalent of approximately 380 cSt.

The fuel system diagram of a 6PC-2-5 Pielstick diesel engine developing 650 hp at 520 rpm is shown in Fig. 16.

The engine *1* is served by the system comprising service tank *13* for diesel fuel, service tank *10* and bunker tank *14* for fuel oil with heating coils, centrifuges *17* and *18* for the cleaning of the diesel fuel and fuel oil, respectively, primary fuel oil filters *8* operating in conjunction with preheater *20* installed upstream of the filters, fine fuel oil filter *22* with automatic discharge of the sediments, viscosity controller *21*, pumps *15* feeding the fuels to the centrifuges, pressure-maintaining valve *2*, straightway valve *3*, fuel leak-off tank *4*, fuel and lube oil leak-off tank *5*, main and standby fuel pumps *6* and *7*, flow meter *9*, deaerator *11*, remotely-controlled selector valve *12*, fuel preheater *16* upstream of the centrifuges and water drain tank *19*.

The centrifuges operate as purifiers, separating both water and sediments from the fuels. The temperature of the fuel oil in the service tanks is maintained not below 313 K (40°C), and the preheater *20* raises the fuel temperature to approximately 363 K (90 °C) before admitting the fuel into the diesel.

25. Lubrication of Internal-Combustion Engines

The oil used to lubricate internal-combustion engines is exposed to the combined effect of high temperature and gases of a varying composition. On trunk-type engines the deteriorating influence is particularly pronounced. The temperature of the bearings and other rubbing surfaces below the piston in the cylinder of this engine is 313-343 K (40-70°C), while the temperature of the cylinder walls at the upper belt is 453-483 (180-210°C), and next to the top piston ring when the piston is at TDC, 463-503 K (190-230°C). Suitable for

use under such conditions is obviously only a crankcase oil with a high flash point.

According to the mechanism of friction, two solid bodies being in contact with, and pressing upon, each other will be in the state of relative displacement or sliding only then when the moving force is higher than the frictional force set up between them. The frictional force varies with the normal load on the rubbing surfaces, their finish and the rate of their relative displacement (Fig. 17). Friction, if set up, causes wear on the components and reduces the engine

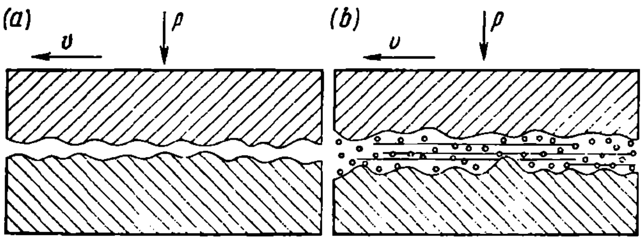


Fig. 17. Modes of friction
(*a*) dry friction; (*b*) friction of complete lubrication

power, because some of it is wasted to overcome friction. The work done by the frictional forces is converted into heat, causing the overheating of mating components which are likely to become seized or even fused in certain cases.

To control the situation, use is made of lubrication which reduces friction and wear. Being fed abundantly and in good time, lubricating oil also cools down the rubbing surfaces, removes the products of wear and provides protection against corrosion at the same time.

Since the rubbing surfaces are bound to operate under various conditions, distinction is made between dry friction, the friction of boundary (or greasy) lubrication and the friction of complete (or viscous) lubrication.

Dry friction occurs when solid surfaces slide over each other with no lubricant being present between them. This type of friction is not permissible, because it may lead to heavy wear and scoring of parts, or even to serious engine breakdowns.

The friction of boundary lubrication results when the film of a lubricant separating the rubbing surfaces is destroyed and patches of dry friction appear, for example, between compression piston rings and the cylinder liner, in the small end bearing of a connecting rod at starting or shutting down the engine, and in any other bearing at low rotational speed and high unit load. Since the wear and

overheating may be very high, steps are to be taken to obtain the friction of complete lubrication.

That type of friction is experienced when the rubbing surfaces are separated by a film of lubricant of adequate thickness. Neither wear nor a loss of energy take place in this case, and the engine operates under optimum conditions.

Adequate lubrication is a factor that determines trouble-free and longer service life of the engine. Therefore it calls for the use of only those grades of lubricating oils which are specified in the engine certificate.

26. Fundamentals of the Hydrodynamic Theory of Lubrication

Let us discuss the way in which a plain bearing operates from the viewpoint of the hydrodynamic theory of lubrication which has been developed by a Russian scientist N.P. Petrov.

The essence of this theory is as follows. When at rest, the journal of a shaft contacts the lower part of the bearing through a thin film of oil particles which adhere to the journal (Fig. 18*a*). As the shaft

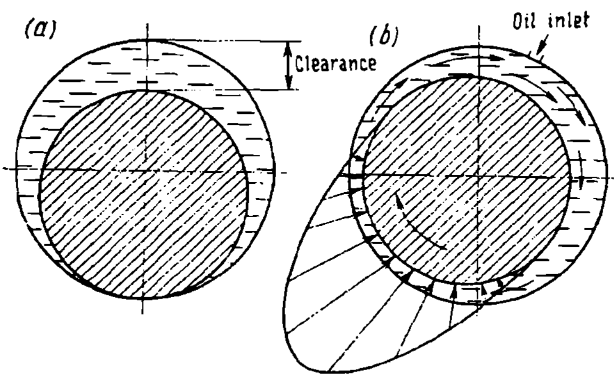


Fig. 18. Journal bearing with perfect lubrication
(*a*) journal at rest; (*b*) oil wedge is created when shaft is rotating

sets out to rotate, a wedge-shaped clearance is formed due to a difference between the diameter of the journal and that of the bearing. The oil adhering to the rotating journal is drawn into the clearance (Fig. 18*b*) and the resulting pressure buildup in the narrowest part of the clearance lifts the shaft clear of the bearing. Subsequently, the clearance increases to a maximum within an arc of 120 deg of the bearing circumference.

Thus, as soon as the shaft rotates with a certain speed, a layer of oil is established between the journal and bearing, keeping them out

of contact with each other. The load on the journal is balanced by the internal pressure in the oil wedge which grows thicker as the shaft gains speed, because more oil enters the clearance due to the journal action.

The friction between oil layers is a function of oil viscosity only, being thus independent of the bearing material and bearing finish. However, it must be borne in mind that, according to the mechanism of hydrodynamic friction, the oil viscosity varies directly with the load and shaft-to-bearing clearance.

When shaft reversals or rapid speed and load changes are likely to be encountered, it is practical to use a more viscous oil than the one used under static conditions. However, excessive viscosity of the oil builds up extra pressure in the oil wedge and is a source of higher frictional losses. These are the points to be taken into account in experimental determinations of the properties of the oil to be used in a newly designed engine.

In a correctly designed bearing, balance between oil viscosity and friction is set up automatically by virtue of the hydrodynamic conditions created when the engine is in operation. In fact, an increase in the angular velocity results in higher friction between oil layers which, consequently, heat up. The resulting increase in the oil temperature diminishes the oil viscosity, thus establishing an equilibrium between friction and the oil temperature.

In a plain bearing, lube oil is fed as a rule into the oil wedge, i.e. into the no-load zone.

27. Properties of Lubricating Oils

The basic properties of lubricating oils are viscosity, coking capacity, sediments content, corrosive effect, base number, flash point, pour point, etc. being as essential as the properties of the fuels used in internal-combustion engines.

Let us look at some of them relating especially to diesel engines.

The viscosity of oil determines the resistance of oil internal cohesive forces and promotes the setting up of certain conditions for the friction between the engine's rubbing components. Oils of a viscosity lower or higher than that of the oil recommended for use on a given engine by its operating instructions, are equally unacceptable. Low-viscosity oil is easily squeezed out through clearances between bearing parts, inviting their rapid wear and overheating that leads to the scoring and runout of inserts. High-viscosity oil adds to the friction and renders the engine starts difficult, especially in cold weather.

Viscosity varies with the temperature: decreasing when the latter increases, and vice versa. The less the viscosity of an oil becomes affected by temperature, the higher is the quality of the oil, for

it is then regarded as being suitable for use within a wide range of temperatures. Stable-viscosity oils find application as lubricants both for highly heated components such as pistons, cylinder liners and small end bearings, and the components operating at low temperatures. This quality of oil is particularly valuable in engine cold starts.

Coking capacity (carbon residue) shows the tendency of oil to form carbon deposits in burning up at elevated temperatures. A high carbon residue leads to the piston rings gumming which deprives them of resilience and ability to move in the grooves. The result is a decrease in the engine power output. The carbon residue varies with the chemical composition of the oil and can be lessened by removing some of the salts at the refinery.

The sediments present in oil consist of the grit resulting from wear and carbon. Their maximum allowable content is 1.5%, or otherwise rapid wear associated with the clogging of oil filters and difficult oil purification will follow.

Corrosive effect relates to the tendency of an oil to oxidize due to the oxygen of air in high-temperature gaseous surroundings. The organic acids—products of oxidation—are detrimental to alloyed bearing metals containing lead.

Base number is one of the most important characteristics of lubricating oils. It indicates the oil's capacity to neutralize the sulphuric compounds which are formed when the engine is run on a sulphur-rich fuel. Acid neutralization prevents corrosion and safeguards engine components against wear.

Flash point defines the oil vapours tendency to ignite. The oil vapour mixture with the oxygen of air in a certain proportion, which may be determined experimentally, is explosive. An explosive mixture of oil vapour and oxygen may also form in a running engine at various temperatures due to a dilution of the crankcase oil with fuel leaks. The amount of diesel fuel present in the crankcase oil can be found from the flash point which has, consequently, changed: the minimum allowable flash point is 443 K (170°C).

The pour point of an oil is to be taken into consideration if the engine is bound to operate at low temperatures. If an oil is to be handled at temperatures exceeding the pour point by 15°C or less, the oil must be preheated.

28. Lubricating Oils Used in Internal-Combustion Engines

Nowadays, some fifty grades of oils are used as diesel engine lubricants in this country. They fall into the following classes: automotive oils, diesel oils, motor oils, aviation oils and special-purpose oils.

Automotive lube oils are marketed, as a rule, with additives, their viscosity at 373 K (100°C) being 6-15 cSt and the pour point being

268 K (—5°C) to 238 K (—35°C). These oils may have the following index marks: АСп6, АСп10, АКзп6, АКзп10, АКп10, АК15, etc. The number in the index shows the oil viscosity in cSt at 373 K (100°C).

Where, A stands for automotive oil; K and C—method of oil refinement, acid or solvent; з—additives for oil thickness; п—multi-purpose additives.

Diesel lube oils are used in some of the high-speed diesels (e.g. Ч 18/22, ЧН 36/45, Ч 23/30, etc.). Their kinematic viscosity at 373 K (100°K) is 8-15.5 cSt and the pour point is 263-248 K (—10 to —25°C). They have the following indexes: Дп8, Дп11, Дп14, and Д11 (Table 2).

Motor lube oils find application on slow-speed diesels operating at not over 600 rpm.

Aviation lube oils are employed in piston-type aircraft engines and some of the high-speed boosted diesels (ЧН 15/18, 12ЧН 18/20 and others).

However, the above classification of lube oils is inconsistent with modern trends, since it does not reflect the requirements to be met by the lubricants with due regard paid to the thermal and dynamic loads, the fuel used on the engine, and other factors.

Therefore, the above classification of lubricating oils has been replaced in this country by a new system which takes into account the service condition of oils. Accordingly, the available range of lube oils has been classed into six groups (А, Б, В, Г, Д, Е) covering several grades which differ from one another by the viscosity.

Group A comprises oils free from additives or containing only oxidants in a minimum amount (additives are chemicals added to lube oil in order to improve its quality). They are used in carburetor engines and in some low-stressed diesels burning low-sulphur fuels.

The oils comprising group Б contain 3-4% of additives and are intended for use in low-stressed diesel engines operating on fuels having 0.2-0.5% sulphur.

Group В covers oils with 4 to 7% of additives used in diesel engines run on diesel fuel with 1% sulphur.

The oils of group Г are used in boosted diesel engines of all applications which operate on low-sulphur fuel (less than 1%); their additive content being 7-12%.

Group Д oils consist of heavily doped varieties containing additives between 18 and 20%. They are used in boosted and highly supercharged engines burning sulphurous fuel (1.5% sulphur).

Group E oils are intended to lubricate the cylinders of large low-speed supercharged diesels operating on high-sulphur residual fuels; the additive content is 25% and upwards.

With the new lubricating oils classification, a new range of lubric-

Table 2

Specifications for Lube Oils, USSR Brands, Used in Diesel Engines

Test	МС-14	МС-20	МК-22	ДП-8 with corrosion inhibitor	Д-11 undoped	ДП-11	ДП-14	ДС-8 (М-8Б) with corrosion inhibitor	ДС-8 (М-8Б) with versatile additives	ДС-11 (М10Б), with corrosion inhibitor
						with corrosion inhibitor				
Kinematic viscosity at 100°C, cST	14 min	20 min	22 min	8-9	10.5-12.5	10.5-12.5	13.5-15.5	8±0.5	8±0.5	11±0.5
Flash point in open cup tester, °C, min	200	225	230	200	200	190	210	190	190	200
Pour point, °C, max	—30	—18	—14	—25	—18	—15	—10	—25	—25	—15
Coking capacity (before doping), %, max	0.45	0.3	0.7	0.2	0.4	0.4	0.55	0.15	0.15	0.30
Acidity, mg KOH/g, max										
when undoped	0.25	0.05	0.1	—	0.15	—	—	0.02	0.02	0.02
when doped	—	—	—	0.10	—	0.10	0.10	—	—	—
Ash, %										
when undoped, max	0.003	0.003	0.004	0.005	0.005	0.005	0.006	0.005	0.005	0.005
when doped	—	—	—	0.25	—	0.25	0.25	0.80	0.70	0.80

ants has appeared in this country and is widely used in marine diesel engines. Worth mentioning are low-alkaline oils employed in the circulation lubrication of high-speed trunk diesels, which are also used as the crankcase oils of crosshead engines and as lubricants of other equipment. Included are also medium-alkaline crankcase oils for high-powered diesels burning fuel with less than 1% sulphur (they are equally good in circulation lubricating systems) and high-alkaline oils for high-powered engines run on fuels with sulphur around 2%.

29. Compounding of Lubricating Oils

Lubricating oils are commonly compounded, using various additives. The main requirements to be met by the additives are effectiveness in minimum amounts irrespectively of temperature and pressure variations in the lubricating system, solubility in the lubricant, long shelf life and low cost. Functionally, the additives may be grouped as follows.

Viscosity improvers serve to raise the viscosity of an oil at high temperatures and prevent its lowering at low temperatures. They are used in an amount 0.5-10% to ease the starting of the engine from cold at low ambient temperatures. A typical viscosity improver is poly-isobutylene, a colourless, sticky and heavy-bodied compound.

Oxidation inhibitors are used to suppress the formation of free radicals in oil which are detrimental to bearing metals alloyed with copper, lead and cadmium. They also have a passivating effect on exposed metal, safeguarding it against acid attacks.

Corrosion inhibitors, added in an amount of 0.5-3%, neutralize the acid materials resulting from burning sulphurous fuels and form strong protective films on the surface of metals. Some of these additives may function as detergents.

Detergents are added to oils in order to clean components from carbon deposits. They also deter deposit formation by keeping in suspension the products of oxidation and insoluble soot particles which are removed eventually from the lubricant by filtration and centrifugation.

Pour-point depressants, added in an amount of 0.1-1%, are used during cold weather operation. They prevent the growth of wax crystals which increase the oil viscosity at low temperatures.

Foam inhibitors suppress the frothing of the oil when this mixes up with air. Frothing impairs lubrication and may set up air locks in the lubricating system. To prevent this, silicons are added in an amount of 0.001-0.0001% which break up the air bubbles and form a thin air-permeable film to keep in check the splashing of the oil.

Sump configuration, foam breakers and the oil quality are other factors that influence frothing.

Widely used in diesel engines are versatile additives producing a combined effect on lubricating oils.

30. Oil Drain Periods

The time when the lubricating oil becomes unsuitable for further use and must be changed varies with a number of factors such as sump capacity (applies to trunk-type diesels), the effectiveness of filtration or centrifugation, engine load, the wear on pistons and cylinders, etc. Ageing with time due to the influence of the oxygen of air, the oil deteriorates: engine gums would form and its chemical composition would change. Therefore, the oil must be replaced at intervals specified in the service manual.

Practical experience goes to show that the maximum sediments rendering the oil unsuitable for further use are 1-1.5%, maximum 3%, if the additives are taken into account. A failure to renew the lubricant at this stage leads to problems in centrifugation and reclamation of oil, and invites rapid wear.

Another criterion of oil rejection is an increase in the acidity to 0.5 mg/g of KOH in the diesels with lead bronze bearing shells and to 2.0 mg/g of KOH in diesels with babbitt-lined bearings. A higher acidity is detrimental to alloyed bearing metals.

Fuel leaks into the crankcase when the engine is in operation produce explosive concentrations of oil vapour and the oxygen of air, reduce the oil viscosity and destroy oil films. Serving as an indicator of oil quality is the flash point which decreases as the lubricant deteriorates. The minimum allowable flash point of a lubricant after a period in service is 443-453 K (170-180°C) against 463-483 K (190-210°C) for fresh diesel lube oils and 473-503 K (200-230°C) for unused aviation lube oils.

No water over 0.5% should be present in a lubricant, whether doped with an alkaline additive or not, to safeguard the engine against corrosion.

The usual practice is to change crankcase oil after 5 000-10 000 running hours in medium-speed diesel engines and after over 10 000 running hours in low-speed ones.

To monitor the lubricant quality, oil test samples are taken every 150 running hours from trunk-type engines and every 500 running hours from crosshead engines, the sample-taking procedure is outlined in relevant instructions. A longer period between the oil tests specified for crosshead engines is accounted for by the fact that in these engines the crankcase oil is less contaminated by the fuel combustion products due to the cylinder bores being separated from the crankcase.

The rated lube oil consumption specified by the manufacturer varies with the type and model of the engine, engine condition and the method of lubrication. An excessive oil consumption betrays wear on the pistons and cylinder lines in general and on the oil-scaper rings in particular.

Normal lube oil consumption in g/kW h (g/hp h) is as follows:

Two-stroke diesel engines	
14Д-100(14 ДН 20.7/2×25.4) . .	3.0 (2.2)
61Б (16 ДРПН 23/2/30)	3.7 (2.72)
8ДКРН 74/160	0.6-0.8 (0.44-0.59)
Four-stroke diesel engines	
12ЧН 18/20	6.0 (4.4)
8Ч 23/30	4.0 (2.94)
6ЧНСП 25/34	3.0 (2.2)
6ЧН 36/45	3.6 (2.64)

The crankcase oil drained from the engine after a period in operation specified in the engine service manual is sent for reclamation.

REVIEW QUESTIONS

1. What is crude petroleum?
2. What is the purpose of the straight-run distillation of petroleum?
3. What are the requirements to be met by fuel of various grades used in marine diesel engines?
4. What are the basic properties of fuels?
5. What fuels are used in marine diesel engines?
6. How is bunkering carried out?
7. Why is it necessary to specify the rated fuel consumption of an engine?
8. What are the peculiarities of diesel engine operation on residual fuels?
9. How are friction between and wear on the rubbing components diminished in a diesel engine?
10. What are the three types of friction and what is the difference between them?
11. Discuss the fundamentals of the hydrodynamic theory of lubrication.
12. What are the basic properties of lubricating oils?
13. Why is the compounding of lubricants practised?
14. Name some additives and characterize them.
15. Why does crankcase oil deteriorate after a period in service?
16. Name the signs which call for the replacement of crankcase oil?
17. What factors influence the oil drain periods?

Chapter V

ENGINE DYNAMICS

31. Forces Acting in a Single-Cylinder Engine and Irregularity of Crankshaft Rotation

It is known that the connecting rod-crank mechanism serves to convert the reciprocating motion of the piston into the rotary motion of the crankshaft.

Let us consider the forces set up in the connecting rod-crank mechanism of a single-cylinder engine. During the expansion stroke, coming on the piston are a force due to the gas pressure and an inertia force of the reciprocating parts. While the former varies with

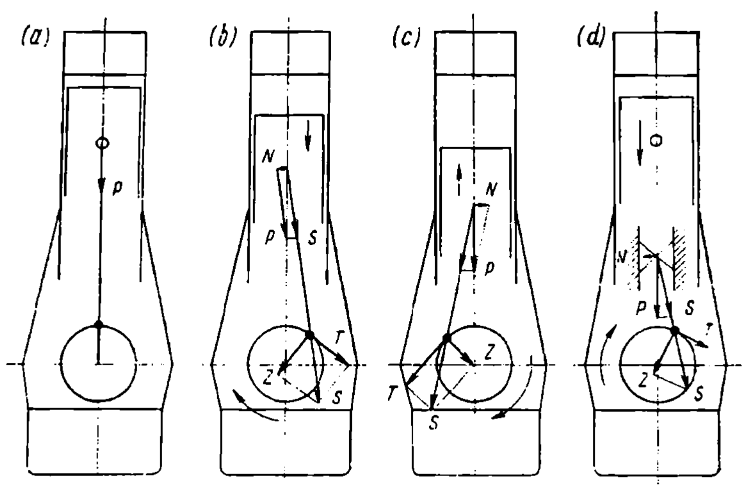


Fig. 19. Forces arising in single-cylinder engine

the crank angle, the latter—equalling the product of the acceleration of the parts and their mass—varies directly with crankshaft speed. The mass of the reciprocating parts equals the mass of the piston assembly and 30-40% of the mass of the connecting rod.

The resultant of the forces, referred to as the motive force P, is applied to the centre of the piston pin, and transmitted to the crankshaft through the connecting rod. The motive force, as this is known from Section 3, is resolved into two components N and S (Fig. 19a and d). The normal component force N presses the piston against the cylinder line in a trunk-type engine or the shoe against

the corresponding guide in a cross-head engine. This force, varying in both direction and magnitude, produces a recurrent piston thrust against the opposite sides of the cylinder liner. It also gives rise to an overturning moment about an arm equal to the distance between the axis of the piston pin and the crankshaft axis. The moment, opposing the direction of crankshaft rotation, is taken up by the bolts holding down the engine to the bedplate.

Bringing the second components force S (see Fig. 19*a-d*) down the line of its action and applying it to the crankpin centre, we resolve it in two components: a force T tangential to the crankpin and a force Z coinciding with the crankpin radius.

The force T produces a torque $M_t = Tr$. Varying with the crank angle from a maximum to a minimum within a certain period, the torque causes the crankshaft to rotate irregularly. The force Z bends the crankpin and creates wear in the bearing.

The crankshaft of a multi-cylinder engine, set to rotate by the torques produced by all the cylinders in succession, will operate more regularly than the crankshaft of the single cylinder engine. However, the torques will not coincide in time, because the cranks are arranged at certain angles to each other rather than in the same plane. This implies that the recurrence of torque alterations increases directly with the number of cylinders and the irregularity of crankshaft rotation decreases.

The work done by a two-stroke engine per each crankshaft revolution, or by a four-stroke one in two crankshaft revolutions, is proportional to an area bounded by the resultant curve of the tangential forces and an abscissa representing the period of the operating cycle. The quotient obtained if this area is divided by the length of the graph gives the value of the average tangential force setting up the engine torque.

Comparing the continuously altering engine torque with the moment due to the force resisting crankshaft rotation, provided this moment is assumed to be constant when the engine is in a steady operating condition, we can see that the torque exceeds the moment at the instants of cylinders' firing and is less than the moment during the intermissions. The two situations are referred to as surplus torque and torque deficiency, respectively, and cause irregular crankshaft rotation.

Defining this irregularity is the so-called irregularity factor of the engine which is the ratio of the difference between the maximum and minimum angular velocities of the crankshaft and the mean angular velocity throughout a cycle of torque alterations. The lower the irregularity factor, the more steadily is the crankshaft rotation.

To minimize the irregularity of crankshaft rotation, use is commonly made of a flywheel fitted to the aft end of the crankshaft. It functions as an accumulator of the energy of the gyrating masses,

storing it at the periods of surplus torque and supplying the stored energy during the periods of torque deficiency. This lessens the irregularity of crankshaft rotation.

Since flywheels carry high inertia loads, they must be manufactured with great care; cracks, pits and other defects are not permissible. A gear ring of the barring gear is cut in the flywheel rim.

Flywheels are not the only means of checking crankshaft rotation, the alternative is to increase the number of cylinders. Diesel engines having more than twelve cylinders commonly dispense with the flywheel, its function being taken care of by the gyrating masses of the connecting rod-crank mechanisms.

The standard irregularity factor of marine diesels is commonly between 1/20 and 1/40; in the case of diesel generator sets, the irregularity factor is 1/150 for a.c. service and 1/300 for d.c. service.

32. Firing Order of Cylinders

To obtain better regular crankshaft rotation, the cylinders of a multi-cylinder engine are arranged to fire in equal intervals. Expressed in terms of the angle on the crankshaft, this interval is $720/z$ on a four-stroke engine and $360/z$ on a two-stroke one, where z is the number of cylinders. Consequently, the crank angles of a four-stroke six-cylinder engine are 120 deg each and those of an eight-cylinder diesel are 90 deg each. In the case of two-stroke engines with the equal number of cylinders as indicated above, the crank angles are 60 deg and 45 deg each, respectively.

However, the above angles do not indicate the way the cranks are arranged in the order of their succession. They are set up with reference to a specific order established with a view to relieving the crankshaft journals between the adjacent cylinders from excessive loads, unavoidable if these cylinders would fire in succession. So, by way of illustration, the cranks of a four-stroke six-cylinder engine are arranged pairwise and unidirectionally in the same plane, the cranks of No. 1 and No. 6 cylinders making an angle of 120 deg with the cranks of Nos. 2, 5 and Nos. 3, 4 cylinders and the two last-named pairs making the same angle with each other.

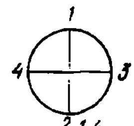
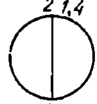
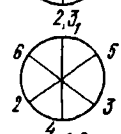
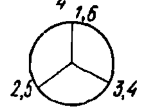
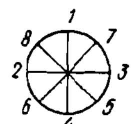
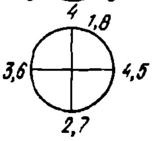
For the crank arrangements and firing orders adopted in various engines depending on the type and number of cylinders see Table 3.

33. Dynamic Balancing of Engines

Coming into play in each cylinder of a running engine is an inertia force of the translating masses of the connecting rod-crank mechanism and an inertia force of the unbalanced gyrating masses. The former is an alternating force in line with the cylinder axis which

Table 3

Firing Order of Two- and Four-Stroke Engines

Engine type	Number of cylinders	Crank angle, deg	Crank arrangement	Firing order
Two-stroke	4	90		1-3-2-4
Four-stroke	4	180		1-2-4-3
Two-stroke	6	60		1-5-3-4-2-6
Four-stroke	6	120		1-3-5-6-4-2
Two-stroke	8	45		1-7-3-5-4-6-2-8
Four-stroke	8	90		1-4-2-6-8-5-7-3

tends either to tear the engine off the foundation or to press it against the foundation by turns depending on the direction of action. The latter acts along the crank web and is constant at any angle on the crankshaft at a given engine speed. It tends either to shift the engine off the foundation, or overturn it. Both these forces induce foundation vibration.

Moreover, the two inertia forces give rise to moments. Those due to the inertia forces of the unbalanced gyrating masses act in the vertical and horizontal planes whereas the moments set up by the inertia forces of the reciprocating masses are confined to the vertical plane. The process of eliminating the unbalanced inertia forces together with the moments they produce is referred to as the dynamic balancing of an engine.

In designing a new engine, a calculation of the unbalanced forces is necessary. The unbalances inertia forces of the gyrating masses are then offset by fitting balance weights to crank webs. The elimination of these forces nullifies the moments they produce. Once each of the cranks has been balanced, all the inertia forces of the gyrating masses of the engine become balanced, as well.

A prudent layout of multi-cylinder engines, i.e. one providing for the right number of cylinders and an appropriate spacing of the cranks apart from each other facilitates the balancing. So, the theory and practice of marine engineering have proved that four-stroke engines with six or more cylinders are at a perfect balance, provided the number of the cylinders is even. This does not apply to two-stroke engines which have always a moment of inertia forces causing an unbalance, no matter how small this moment may be. On the other hand, two-stroke engines materially add to their balance with an increase in the number of the cylinders. Not infrequent are the cases when a two-stroke engine with an odd number of cylinders appears to be better balanced than a four-stroke engine having the even number of cylinders. This is why two-stroke engines with seven or nine cylinders enjoy wide-spread application.

It is a frequent occurrence that balanced engines begin to vibrate at starting, the vibration gradually dies out as more cylinders develop own power and the engine speed increases. The background of this phenomenon is intermittent fuel delivery and, as a result, misfiring of some cylinders gives rise to unbalanced inertia forces and moments. As the engine warms up, the combustion pressure in the cylinders levels up and the unbalance grows lower.

34. Torsional Crankshaft Vibration and Critical Engine Speed

The crankshaft of a main propulsion plant along with the flywheelgears and various elements of the propeller shafting form an elastic system incapable of being absolutely stiff. Therefore, a torque applied to the crankshaft causes it to twist within the elastic limits.

When the torque is removed or lessened, the crankshaft responds by untwisting and twisting in the opposite direction on passing through a state of equilibrium. This state will recur, for the crankshaft will be urged by the elastic forces of its material and the inertia forces of its masses to vibrate at a certain frequency.

The vibration brought about by the elastic forces of crankshaft material and the inertia of its masses in the absence of external forces is referred to as free or natural vibration. The vibration of the crankshaft and the shafting coupled to it which is induced by a variable engine torque is termed forced vibration.

The relative vibration of the masses of the elastic system causing its segments to regularly twist and untwist is known as torsional

vibration. The coincidence of frequency of the natural vibration and frequency of the forced vibration is referred to as resonance. Causing vibration and local overheating of the shafting, it brings about an overstressing so high that a breakdown may occur.

A rate of crankshaft rotation inviting resonance is termed critical speed. Inherent in an engine may be more than one critical speed leading to resonance. A critical speed, manifesting itself by a sharp increase in the amplitude of torsional shaft vibration, is hazardous for the engine. However, it must be passed—the sooner the better, which is a practice the personnel is instructed to follow. The range of engine speeds immediately below and above the critical speed is termed barred zone. It is specified in the engine log book.

A critical speed can be determined with the aid of an instrument called torsiograph which automatically records the frequency and amplitude of a torsional vibration on a paper tape. In operation, a critical speed causes the crankshaft's overheating, the pinking and vibration of the engine. To avoid resonant frequencies of vibration, the remedy is to set the barred zone clear of the operating range of engine speeds. This can be done by adjusting either the rate of crankshaft rotation, or the mass of the flywheel or the firing order of the cylinders. However, the most effective means of reducing the amplitude of torsional vibration is the sectionalizing of the shafting and interposing special couplings between the sections. The same effect can be obtained with the aid of vibration absorbers which are fitted to the crankshaft to dissipate the energy of vibration in a given range of engine speeds.

35. Torsional Vibration Absorbers

Marine diesel engines are fitted with torsional vibration absorbers of the mechanical or hydraulic type which convert the energy of torsional vibration into heat energy dissipated into the surroundings.

A mechanical vibration absorber relies for operation on the friction set up between its rim and the crankshaft, in fact either fore or aft section of the shaft. Referring to Fig. 20, a rim *5* of the vibration absorber accommodating spring packs *4* is a sliding fit on a hub *1*. Each spring pack made up of fourteen or more spiral or U-shaped laminated springs inserted into one another fits into a recess extending from the rim into the hub over one half of its depth. Retainers *3* prevent the spring packs from rotation, and flanges at the sides of the rim counteract axial displacement. All the above parts are contained in a housing *2*.

In service, the hub *1* vibrates integrally with the crankshaft whereas the rim *5* is reluctant to change its angular velocity by virtue of inertia. The difference between the angular velocity of the rim and that of the hub brings about a deformation of the spring packs.

This facilitates a damping of the energy of the vibration and a lessening of its amplitude.

The vibration absorber also prevents the possibility of resonance, for the angular velocity of the hub is invariably at variance with its mean value due to the torsional vibration so that no coincidence

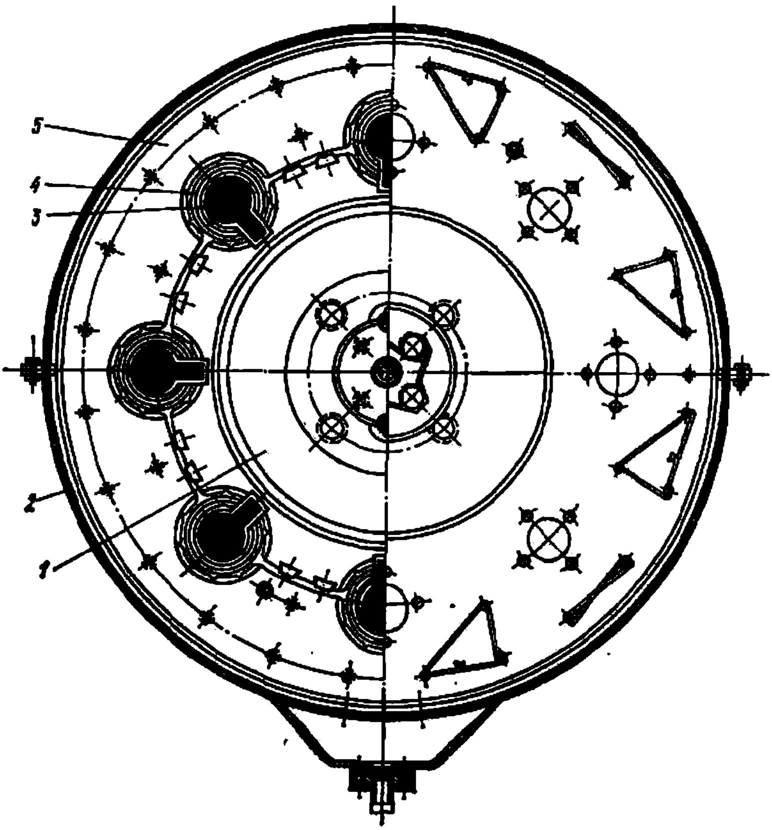


Fig. 20. Torsion vibration absorber of spring type

of the frequencies of the free and forced vibrations is likely to occur.

Some marine propulsion installations use vibration absorbers of the pendulum type. The hub of the absorber is rigidly attached to a crankshaft end and the counterweights are loosely fitted to the absorber body by pins. Any alteration in the crankshaft speed induces an oscillatory motion of the pendulums at a certain frequency. The weight of the pendulum is either calculated, or determined

experimentally so that their vibration adding up with that of the shaft lessens the torsional crankshaft vibration.

In a hydraulic vibration absorber, a fluid contained between the absorber body and the hub substitutes for the springs. The fluid is commonly a silicone, a solution of silicon salts, characterized by high viscosity, thermal and oxidation stability. Responding to the difference between the angular velocities of the rim and the hub resulting from torsional vibration, the fluid absorbs vibration owing to its high viscosity. Silicone vibration absorbers are simple and dependable.

REVIEW QUESTIONS

1. What forces act on a crankpin in operation?
2. How is the regular crankshaft rotation ensured in multi-cylinder diesel engines?
3. What means are used on diesel engines to balance the inertia forces of gyrating masses?
4. What vibration of the crankshaft is termed free vibration?
5. What vibration of the crankshaft is referred to as forced vibration?
6. What is the resonance of vibrations?
7. How can torsional vibration be suppressed or reduced?

Chapter VI

THE ENGINE STRUCTURE

36. The Bedplate

The bedplate is a base which carries the rest of the elements of the engine structure. Taking the dead loads of the supported components, the inertia loads of the moving parts, and the forces arising due to the gas pressure in the cylinders, the bedplate must be of a strength and stiffness, both in the fore-and-aft direction and athwartships, which will ensure normal functioning of the crankshaft.

Referring to Fig. 21, the bedplate is made up of two longitudinal girders *1*, forming the side walls, and a set of transverse I-beams or box girders *2* strengthened with stiffeners. The transverse strength members give support to the main bearings which are accommodated in seats *3* provided in the top flanges. In some diesel engines, transverse members are provided with bosses passing through which are the through bolts.

The bedplates of medium-powered and low-speed high-powered diesel engines are castings. Their structural material is cast iron

with an ultimate strength in tension and bending of the order 18-28 and 36-48 kg/mm², respectively. In order to promote product unification and cut both weight and manufacturing cost of diesel engines, modern practice has made recourse to welded bedplates; structural materials used are steel plate and stamped strength members of appropriate configuration. Welded bedplates are lighter than cast ones by 25-35%.

Bedplates of high-speed engines are made from silumin, an aluminium alloy added to which is silicon.

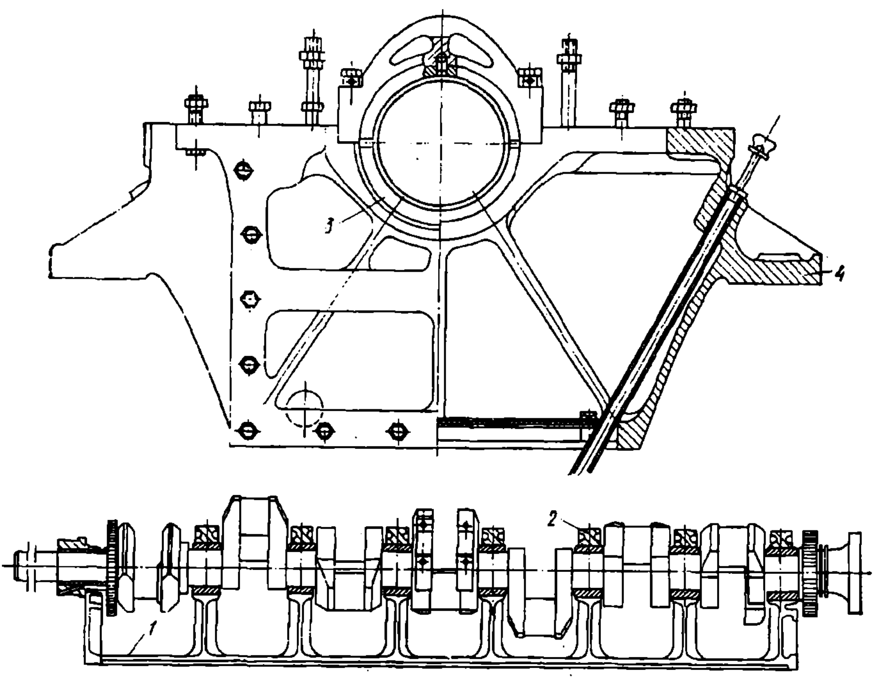


Fig. 21. Bedplate of engine

Seating flanges *4 extending* all the way along the bedplate in its lower part serve to attach the engine to the hull foundation. At least 25% of the bolt holes in the seating flanges must be reamed to accommodate dowel bolts ensuring accurate fit up of the bedplate and the hull foundation.

The underside of the bedplate is adapted to receive the oil pan (or oil sump) of welded construction which is attached to the longitudinal girders by bolts or studs (Fig. 22). The oil pan is inclined towards the central or aft part of the bedplate in order to facilitate the flow

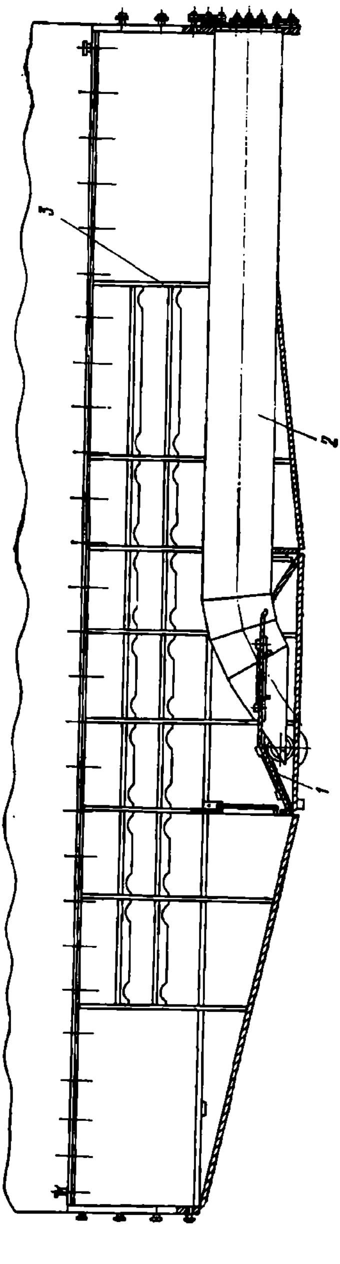


Fig. 22. Oil pan

of the oil dripping from the bearing or pistons (applies to oil-cooled engines) and leaving into an oil collecting tank or suction line *2* of the oil pump. A strainer *1* is fitted between transverse partitions *3* of the oil pan to minimize the frothing of oil in service and prevent foreign articles from getting into the lubricating system.

37. Main Bearings

The main bearings contained in seats, which are cut in the transverse strength members of the bedplate, serve to give support to the crankshaft. They consist of an upper shell *1* and a lower shell *2* (Fig. 23) held fast by a cap secured to the bedplate by studs or special jacks.

In service, the main bearings sustain loads due to the gas pressure in the cylinders and the inertia of gyrating masses of the connecting-rod crank mechanism which alter in direction and magnitude. The resulting unit bearing pressure may be $1\,370 \times 10^4$-$1\,960 \times 10^4$ N/m^2 (140-200 kgf/cm^2) on low speed engines and $3\,430 \times 10^4$-$3\,680 \times 10^4$ N/m^2 (350-375 kgf/cm^2) on high-speed ones. To stand this pressure and ensure trouble-free operation of the engine, the bearings must have high wear resistance and strength. Meeting these requirements are bearing shells made from low-carbon steel (0.1-0.3 %C) or grey cast iron with an ultimate strength in tension and bending of 21-40 and 24-44 kg/mm^2, respectively. The bear-

ing surface of the shells is provided with a lining.

Most commonly used in marine applications as the lining of main, crankpin and end journal bearings is babbitt, a lead-base alloy. Characterized by good ductility and grinding-in, babbitt readily softens, should an area of dry friction come into existence, and sags so that the load becomes uniformly distributed over the entire bearing surface of the shell. Other assets of babbitt are high thermal conductivity, good antifriction properties and embedability, i.e. the ability to embed foreign particles in itself, safeguarding the crankshaft journals and crankpins against scratching and scoring.

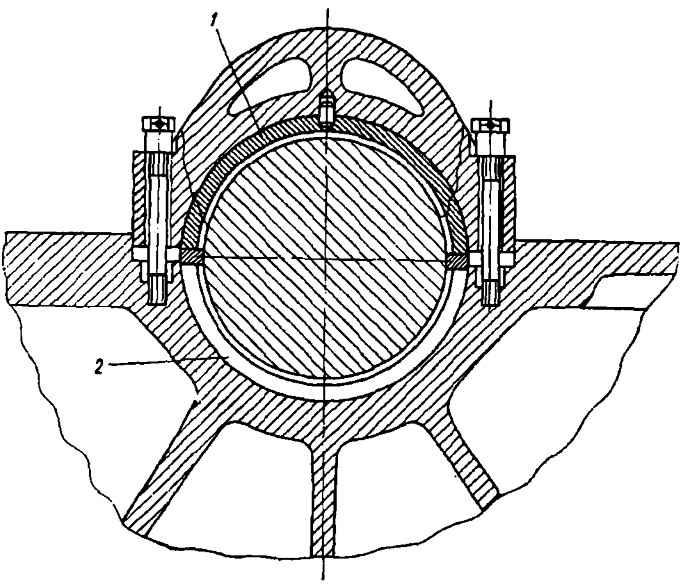


Fig. 23. Main bearing

In use in this country is babbitt containing 82.5-84.5% of tin, 10-12% of antimony, and 5.5-6.5% of copper. The thickness of the babbitt lining varies between 5 and 20 mm depending on the journal or crankpin diameter. However, low fatigue strength of babbitt permits its use for bearing pressures of $1\,760 \times 10^4$-$2\,450 \times 10^4$ N/m^2 (180-250 kgf/cm^2) at temperatures not over 373 K (100°C), or otherwise the flaking and pounding out of the lining may occur.

In applications where the bearing pressure is $2\,450 \times 10^4$-$3\,430 \times 10^4$ N/m^2 (250-350 kgf/cm^2) and the temperature is 200°C, use is made of main bearing shells lined with a 0.7- to 5-mm, depending on the bearing diameter, layers of leaded bronze (30% of lead and 70% of copper). However, this bearing metal is less ductile than

babbitt and, as a result, poorly ground-in. Dry friction may cause failure of leaded bronze bearings leading to a breakdown of the

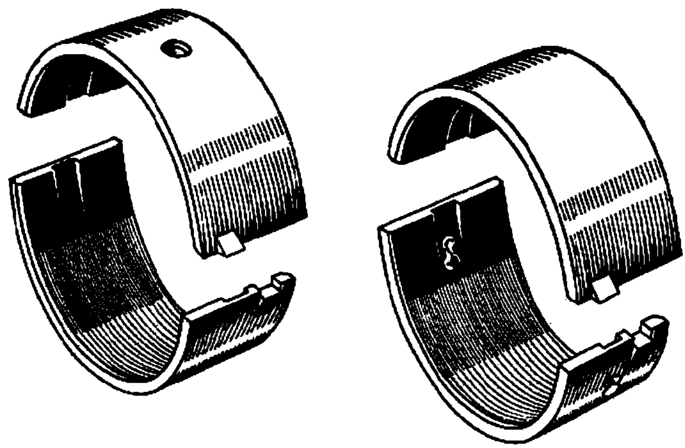


Fig. 24. Main bearing shells

engine. To prevent this, leaded bronze shells require very accurate machining. Their low corrosion resistance calls for using low-acidity lubricating oils which must be replaced at frequent intervals. To ensure satisfactory running against leaded bronze shells displaying significant hardness, the journals and crankpins are either induction hardened or nitrided.

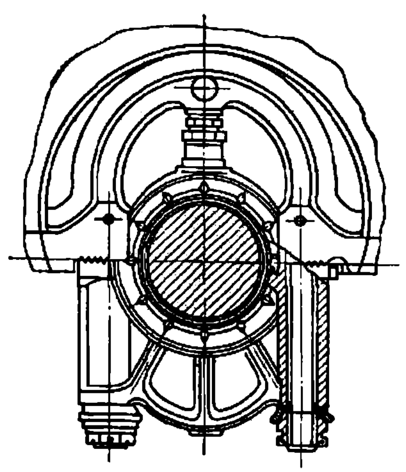


Fig. 25. Main bearing cap held fast by jack

In engines where the bearing pressure is not over 980.66×10^4 N/m^2 (100 kgf/cm^2), the main bearings are lined with an aluminium-base alloy applied in the form of a bimetallic strip.

The running clearance of babbitt-lined bearings is adjusted by brass shims placed between the shells at the joints. No shims can be used in leaded bronze-lined bearings.

Heating up in operation to 313-323 K (40-50°C), the crankshaft elongates by 1-3 mm. To accommodate this increase in the crankshaft length, the main bearing shells are made slightly longer than the

shaft journals. The axial load due to the axial displacement of the crankshaft is taken up by one of the main bearings which is provided in the form of a thrust bearing. The thrust loads produced by the propeller are supported by a special thrust collar which the line shafting is provided with.

The main bearings are lubricated by oil fed from the oil gallery and introduced into the bearings in the unloaded sectors, i.e. at the top and a side surface of each journal. Oil grooves (Fig. 24) of a depth axially decreasing to zero are provided next the shell joints to distribute the oil around the crankshaft.

Each main bearing cap is held down by studs or by a jack abutting against the cap with one end and against a thrust disc of the cylinder block with the other one, using a hardened button-head bolt (Fig. 25).

38. The Crankcase

The crankcase rests on the bedplate, linking it to the cylinder block, and forms a totally enclosed space. In service, the crankcase is in the state of combined stress in bending and tension due to the normal force, maximum combustion pressure and the unbalanced inertia forces of the moving components. Obviously enough, the crankcase must have adequate strength and stiffness combined with low weight.

Depending on engine model, use is made of barrel-type and split-type crankcases of welded or cast construction.

High-speed engines employ crankcases cast from aluminium-base alloys or fabricated from steel plate, the type of the crankcase being decided by engine power output. Some of the low-speed crosshead diesels make use of crankcases consisting of A-frames (Fig. 26) covered with steel sheets with manholes for the inspection of and access to the propulsion mechanism. Bolted on the inside of A-frames are cast-iron guide bars which support the reciprocating crosshead and take the normal loads set up by the connecting rod-crank mechanism. However, the I-beam or box-section A-frames are rather massive, irrespectively of the structural material which may be cast iron or steel plate, and are ousted from marine application. Modern practice favours the single crankcase for all the cylinders.

To relieve the crankcase from tensile loads, use is made of steel through bolts (Fig. 27) tieing together the bedplate, crankcase, cylinders and, sometimes, the cylinder head. Since the through bolt tightening force exceeds the maximum force due to the combustion pressure by 25-50%, the engine structure work in compression and tensile load are taken by the through bolts. Their material is medium-carbon steels (0.35-0.40% C) or alloyed steels (0.18-0.20% C; Cr and Ni content less than 1.5%).

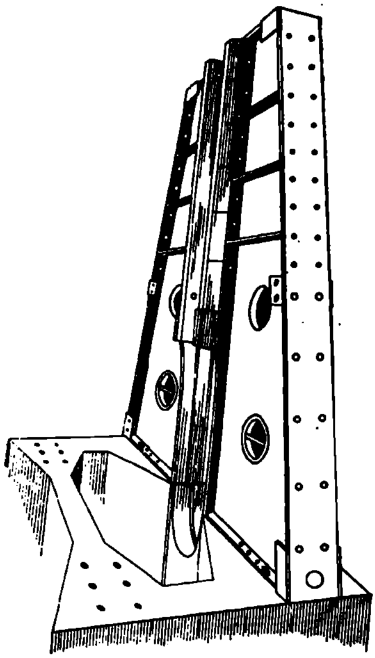


Fig. 26. A-frame of crankcase

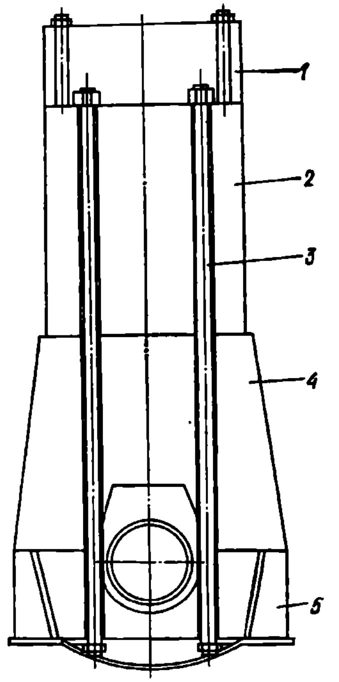


Fig. 27. Through bolts
1—cylinder head; 2—engine cylinder; 3—through bolt; 4—cylinder block; 5—bedplate

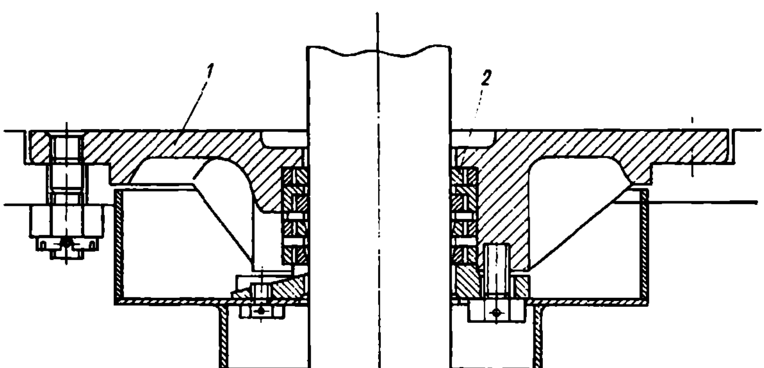


Fig. 28. Piston rod gland
1—diaphragm; 2—gland

To prevent the oil vapour formed in the crankcase from exploding, should a spark enter the crankcase during a combustion blow-by, each of the cylinders of large crosshead engines is provided with a diaphragm and a piston-rod gland (Fig. 28). This arrangement extends oil replacement periods, because there occurs no crankcase oil contamination by the exhaust gases no matter how long the engine is in operation. It improves the dependability of the engine and extends its service life though it complicates the design and invites difficulties in erecting the engine.

39. The Cylinder Block

A practice which is resorted to in trunk-type engines in order to obtain a compact and light-weight engine structure is to cast the crankcase integrally with the cylinder block (Fig. 29).

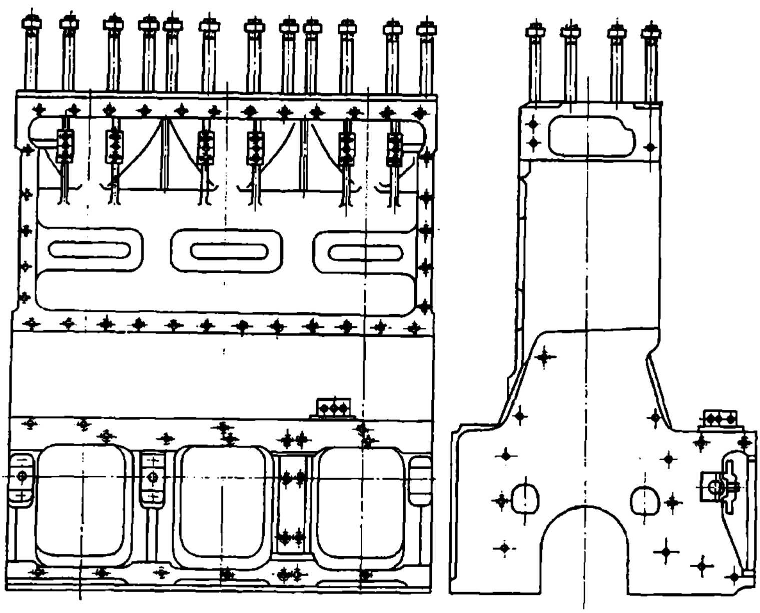


Fig. 29. Twin-section cylinder block of two-stroke diesel engine

The engine blocks may be cast and welded in one-piece or of composite construction, with the individual sections being bolted or studded to each other. So, for example, the four-, six- and eight-cylinder engine blocks of the ДР 30/50 engine consist of a four-cylinder section, two three-cylinder sections and two four-cylinder sections, respectively. This arrangement provides economy in

maintenance cost, for it is possible to remove a section with a defective cylinder and replace it with a new section instead of renewing the entire engine block.

The material of engine blocks is grey cast iron with an ultimate strength in tension and bending of 18 and 36 kg/mm^2, respectively. Also in use is special cast iron with a tensile strength of 45 kg/mm^2. The engine blocks of high-speed medium-powered engines are made from silumin alloys.

The cylinder block is attached to the upper half of the crankcase by studs. Cylinder liners are press-fitted into the cylinder block. Once in place, they build together with the cylinder block a space referred to as water jacket.

The engines arranged for the withdrawal of the pistons downward from the cylinders on the removal of the main bearing caps, are characterized by an extra dependability because the crankshaft journals and crankpins can be made of a diameter which is larger than on those engines which are adapted for an upward removal of pistons through the cylinder bore. Cylinder blocks of this kind are commonly made integrally with the upper half of the crankcase and are referred to as engine structures with the underslung crankshaft. They are widely used on many diesel engines made in this country (12ЧН 18/20, 16ДРПН 23/2 × 30) and abroad.

40. Cylinder Liners

Cylinder liners belong to most essential and crucial diesel engine components. In conjunction with the pistons and cylinder heads they form spaces in which fuel combustion takes place and the energy thus liberated is converted into the reciprocating motion of the pistons.

In reciprocating, a piston of a trunk-type engine gives rise to considerable frictional forces causing wear of the cylinder liner. It is also deformed by a normal component of the gas pressure which is applied through the trunk. This component depends on the maximum combustion pressure p_z and the cylinder bore, varying between $0.1p_z$ and $0.12p_z$. Finally, the cylinder liner sustains significant thermal deformations and is subjected to acid-induced corrosion. Apparently, to stand the combined effect of these detrimental factors, the material of cylinder liners must meet stringent strength requirements. Commonly in use are cast irons with a tensile strength of 24-32 kg/mm^2 and a bending strength of 44-52 kg/mm^2 due to alloying with nickel, chromium, titanium, vanadium, etc.

Alternatively, cylinder liners are made from low-alloy steel (0.35% C; Cr, Mn and other additives 1.5% or less), their bores being honed, coated with porous chromium or nitrided for extra wear

resistance. A helical pattern of grooves some 0.04 or 0.05 mm deep is provided in the liner bores to retain the oil film (Sulzer engines). From the outside, cylinder liners are given a corrosion-inhibiting coating, using, for example, phenol-formaldehyde lacquer. Also in use is anodizing in an electrolytic bath producing a film of oxide on the surface, as this is the case in the high-speed, four-stroke ЧН 18/20 marine diesel.

Circulated through the water jacket to cool the cylinder liners is commonly fresh water with corrosion-inhibiting additives. If outboard water is used, sacrifical zinc anodes must be fitted to the liners. Since the emf of zinc is higher than that of iron, the former will be destroyed by electrochemical corrosion while the latter will stay intact.

The liners in direct contact with the cooling water are referred to as "wet" liners. "Dry" liners are also used but in diesels with a cylinder bore not over 200 mm.

The cylinder liners of two-stroke engines are provided with exhaust and scavenging ports (Fig. 30), their size and location being decided by the system of scavenging used. To prevent an ingress of cooling water into the cylinder, the port belt is sealed off by rubber rings *7*. In the area of high temperatures next the scavenging ports *5* and the exhaust ports *4*, the sealing is effected with copper rings *6* which are rolled into the corresponding grooves and turned then in a lathe.

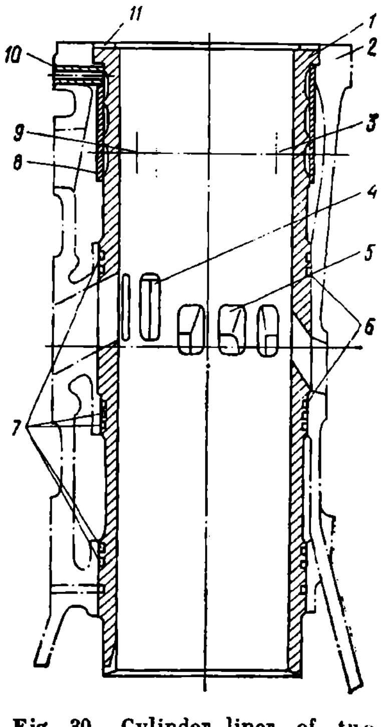


Fig. 30. Cylinder liner of two-stroke diesel engine

Each cylinder liner is press-fitted into the respective bore of the cylinder block *2* so that its top flange *1* bears upon the block through a coat of thick white lead paint or an annealed copper gasket. Prior to fitting the liners, their locating collars are lapped to the appropriate mounting surfaces of the cylinder block. The lower end of each liner is free to expand axially when the liner heats up. An annealed copper gasket *11* fitted into an annular groove in the top surface of the liner flange and held fast by the cylinder head prevents compression blow-by.

The water which cools the liners enters the jacket in its lower part and, on rising, is admitted into the cylinder head over connecting pipes *10*, as this is the case on the engines of the Д and ДР 30/50 ranges. A labyrinth packing encased with a shroud *8* may be fitted to the liner. The passages of restricted cross-sectional area which are formed between the external liner ribs and shroud accelerate the flow rate of the cooling water and ensure an efficient cooling of the hottest topmost part of the liner.

The liner bores are lubricated with the oil fed through lubricating holes *3* and *9* connected to corresponding oil lines. Water holes provided in the cylinders of the Д and ДР 30/50 diesel engines betray leaky seals or rubber rings.

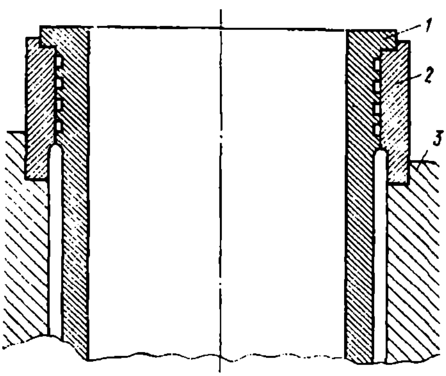


Fig. 31. Cylinder liner supported by cold outside ring

1—cylinder liner; *2*—cold outside ring; *3*—cylinder block

In the Sulzer RD-90 diesel engine, a cold outside ring *2* (Fig. 31) is fitted to the liner *1* with a clearance between the liner flange and cylinder block. In operation, the liner expands due to elevated temperature by an amount greater than the clearance separating it from the ring with the result that the ring takes the stresses set up by gas pressure and thermal loads and promotes the cooling of the liner and topmost piston rings. A reduction in the engine height and, consequently, in the mass of the engine is another asset of the cold outside ring. A steel fire ring is provided inside the liner to give protection against impinging fuel jets.

The bore of the cylinder liners in high-speed trunk-type engines is lubricated by the oil dripping from the crank and connecting rod swinging back and forth along a complex path. In two-stroke engines the oil vapour entering the liner bore together with scavenging air may lead to abundant lubrication. To remove surplus oil from the liner walls, the pistons of these engines are fitted with more oil-scraper rings than common four-stroke diesels. Sulzer and MAN engines dispense with oil-scraper rings.

Low-speed engines are pressure-lubricated by the oil admitted into the cylinders through the pipe unions screwed into the cylinder walls. The oil is supplied under a pressure of 980×10^4 N/m^2 (100 kgf/cm^2) by means of lubricating plunger-type pumps driven by the camshaft or piston motion. The squirting takes place at a rate

of 10-12 drops per minute from each union at the instant the piston is at TDC. There is a total of four unions located pairwise at the camshaft and exhaust sides, level with the second and third piston rings.

The non-uniform heating of the liners in two-stroke engines, which is unavoidable due to the difference in liner temperature next to the scavenging ports and at the exhaust ports, gives rise to non-uniform liner deformation distorting the shape of the liner bore. To avoid this, use is made of split liners. On some larger slow-speed diesel engines the problem is taken care of by using built-up liners fitted along a zigzag or straight line with a clearance of 2-2.5 mm between the two liner halves. A collar at the bottom of the upper half fitting into a groove at the top of the lower half ensures alignment.

To alleviate the thermal stresses and the resulting axial strains, the MAN KZ 86/160 diesel engine employs built-up cylinder liners. The upper half of each liner is pierced with the scavenging and exhaust ports and the lower half serves as the guide for the piston, and is held fast to the upper half by means of a flanged joint.

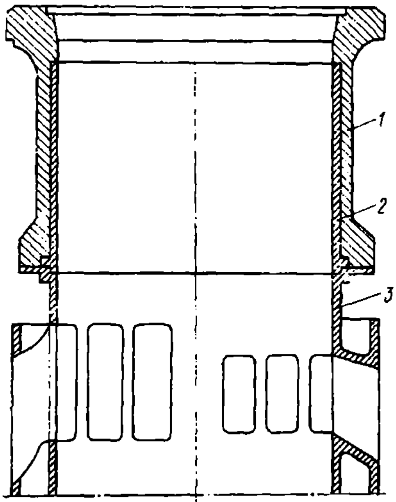


Fig. 32. Cylinder liner of built-up construction

Built-up cylinder liners are also used on the "Fiat" 900S diesel engine (Fig. 32). The upper part of such a liner consists of a cast iron sleeve *2* press-fitted into a steel liner *1*, the lower half *3* of the liner is in cast iron. Renewed after a period in service is only the press-fitted cast iron piece and not the entire sleeve.

To keep the scavenging and exhaust ports clear of carbon deposits, the ribs between the ports need effective cooling. This is ensured by circulating jacket water through drillings in the ribs between the exhaust ports (diesel engines of the Д and ДР 30/50 types).

41. Cylinder Heads

The cylinder head provides a cover for the cylinder liner, and its lower deck together with the upper liner belt and piston form the combustion chamber. In operation, the cylinder head sustains signi-

ficant dynamic and thermal loads caused by the combustion pressure and high temperature.

The cylinder head is made up of an upper deck and a lower deck. The former is fitted with inlet and exhaust valves (in four-stroke engines), a fuel injector, a starting air valve, a safety valve and an indicator cock. The latter contacts the exhaust gases, being the hottest part of the cylinder head. The decks are interconnected by spacer plates forming intricate passages. On engines of the divided-chamber type, the cylinder head also accommodates either the turbulence chamber or the pre-chamber. To minimize the heat stresses, the cylinder head is cooled by the water admitted from the cylinder block water jacket via the connecting pipes.

The studs holding down the cylinder head limit its thermal expansion. The stud nuts are to be tightened uniformly in a certain order, usually in two-three rounds, using a torque-indicating wrench. However, the combined effect of mechanical deformations, and non-uniform heating and cooling results in metal overstress and minute cracks in the lower deck of the cylinder head. The cracked material, being susceptible to the corrosion by virtue of the exhaust gases, increases in volume (the volume of an oxide is roughly three times that of pure metal) and develops more cracks. This means that the metal of the cylinder head must display good casting characteristics, high strength, thermal and corrosive resistance. These requirements are met by grey cast irons with a tensile strength of 24-32 kg/mm^2 and a bending strength of 44-52 kg/mm^2 and by a high-temperature molybdenum steel (0.3% C, Mo 1.5% or less) which is used as the material of the lower deck.

Some of the light-weight high-speed engines are provided with cylinder heads made from aluminium alloys which, as a rule, cover either the entire length of the cylinder block, or two, three and more cylinders.

On some high-speed four-stroke diesel engines the cylinder head is provided with two inlet valves and two exhaust valves, the flow section of the former being sometimes twice that of the latter. The result is better clearing of the cylinders from the burnt gases and a higher coefficient of admission.

The cylinder head of a two-stroke diesel engine (Fig. 33) is of a less elaborate design than the one of the four-stroke engine, because most of the two-strokes dispense with the inlet and exhaust valves (this does not apply to uniflow-scavenging two-stroke diesels which feature exhaust valves).

Multi-cylinder engines are fitted with cylinder heads which can be interchanged. Individual cylinder heads are more practical than multiple ones, for they are not only less difficult in manufacture and erection but also more economical: an individual cylinder liner costs less to replace than a multiple one. Therefore, multiple cylinder

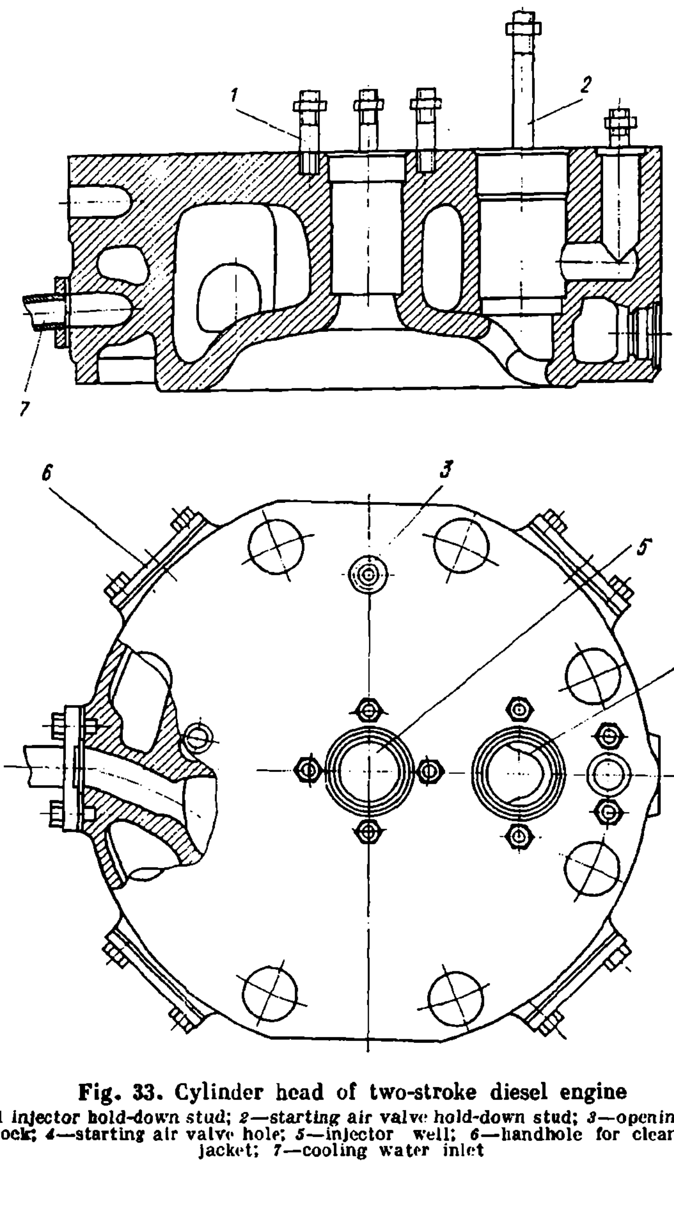


Fig. 33. Cylinder head of two-stroke diesel engine

1—fuel injector hold-down stud; *2*—starting air valve hold-down stud; *3*—opening for indicator cock; *4*—starting air valve hole; *5*—injector well; *6*—handhole for cleaning water jacket; *7*—cooling water inlet

heads are commonly used on smaller engines with a cylinder bore of 200 mm and less, and also in those cases where a low-weight engine structure needs extra ruggedness.

In order to lessen thermal stresses some engines (Sulzer RD-90 and RND-105, MAN KZ 70/120 and KZ 86/160) use cylinder heads split into two concentric parts. A ring-shaped outside part *1* (Fig. 34) sustains high mechanical stresses and is therefore cast from steel. Fitting into the outside part is an insert *2* which takes the combustion pressure only. The insert is fitted with a fuel injector, an air starting valve, a relief valve and an indicator cock.

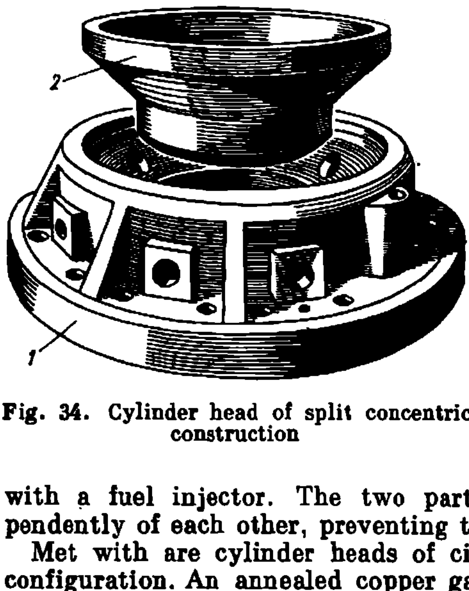


Fig. 34. Cylinder head of split concentric construction

The concentric cylinder head of MAN diesel engine consists of a toroidal lower part, made of steel, which is fitted with a starting air valve, a safety valve and an indicator cock and supports an upper part with a fuel injector. The two parts expand in operation independently of each other, preventing the setting up of high stresses.

Met with are cylinder heads of circular, square and polygonal configuration. An annealed copper gasket is commonly placed between the cylinder head and cylinder block.

Sound cylinder head design is not the only prerequisite of trouble-free operation. Correct running of the engine is another factor of paramount importance. So, for example, carrying full load without heating up the engine, overloads, and filling up the cooling system with a very cold water may cause cracking and failure of the cylinder head.

REVIEW QUESTIONS

1. What are the components of an engine structure?
2. What purpose does the bedplate of an engine serve?
3. What is the function of main bearings?
4. Describe main bearing shells.
5. What purpose does a cylinder liner serve?
6. What purpose does a cylinder head serve?

Chapter VII

THE CONNECTING ROD-CRANK MECHANISM

42. Connecting Rod-Crank Mechanisms of Trunk-Type and Crosshead Engines

The connecting rod-crank mechanism serves to convert the reciprocating motion of the piston into the rotary motion of the crankshaft. In operation it takes loads which vary with the angle on the crankshaft and can be determined with appropriate mathematical relations. Referring to Fig. 19, the motive force coming on the piston can be resolved, using the parallelogram of forces, into a number of components of which only one is of utility, *viz.* P_t, producing the crankshaft torque. The composition of the mechanism varies with the trunk and crosshead engines.

Most of the medium- and high-speed four-stroke diesels are of the trunk type. High-power output low-speed marine diesels (predominantly two-stroke ones) with a cylinder bore of 400 mm and upwards are of the crosshead type. The cylinder of a crosshead engine is relieved of the normal loads by virtue of the crosshead which are transmitted to the guide bars via the reciprocating slides. Proper alignment of the piston in the cylinder makes it operate without direct contact with the cylinder liner.

This feature of the design adds to the dependability of crosshead engines. It is also an asset that high-output models of these engines are provided with piston-rod glands separating the cylinder bores from the crankcase interior. This arrangement prevents the ingress of exhaust gases into the crankcase, thus safeguarding the crankcase oil against contamination with carbon and other products of fuel combustion. Yet, the crosshead substantially adds to the height and, consequently, to the mass of the engine. Therefore, as far as smaller vessels are concerned, the trunk engine has there no alternative.

43. The Piston

Taking the gas pressure during the working stroke of the engine, the piston (Fig. 35) transmits it to the crankpin by way of the connecting rod. In cooperation with the cylinder liner and cylinder head, the piston also forms the combustion chamber, sealing off the cylinder interior and preventing combustion blow-by into the crankcase at the same time. In two-stroke diesel engines, the piston covers and uncovers the scavenging and exhaust ports.

The piston of a trunk engine consists of crown *4*, skirt *2*, piston pin *3* linking the piston to the small end of the connecting rod, bosses supporting the piston pin, compression rings *5* and oil-scraper rings *1*. Ribs *6* provided inside the piston increase its strength and contribute to the cooling of the crown.

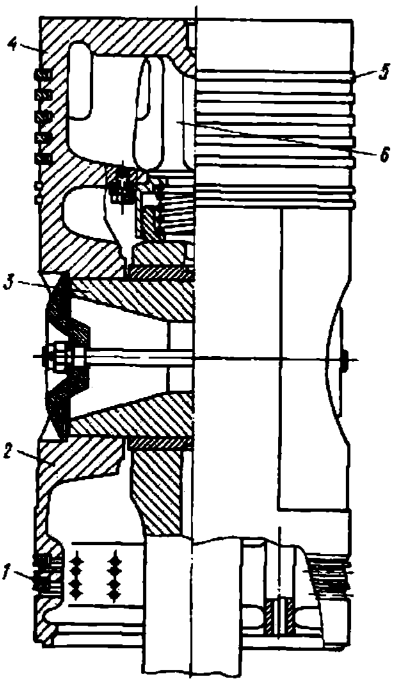


Fig. 35. Piston of engine cylinder

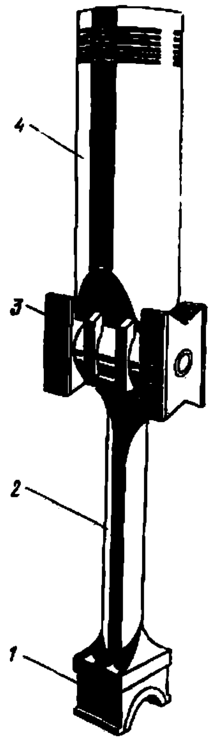


Fig. 36. Connecting rod-crank mechanism without piston rod

1—big end of connecting rod; *2*—connecting rod; *3* — crosshead; *4*—piston

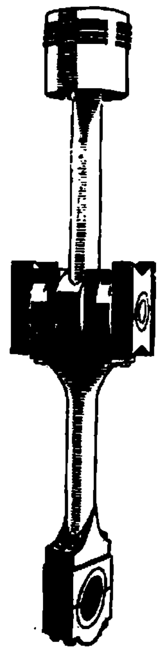


Fig. 37. Connecting rod-crank mechanism with piston rod

Referring to Fig. 36 which depicts the connecting rod-crank mechanism of the Sulzer RD-76 crosshead diesel engine, the trunk piston employed there dispenses with its rod. Pistons of other crosshead engines feature a much shorter skirt alongside with a steel crown which are attached to the top flange of the piston rod (Fig. 37). Short pistons obviously reduce the height of the engine.

The piston would be apparently incapable of reciprocating inside the cylinder liner unless an appropriate clearance separates it from the liner walls. An optimum cylinder clearance is determined by a number of factors such as the temperature of the piston and that of the liner, the cooling effect obtained, the method of lubrication, structural material, cylinder bore, etc. The reciprocating piston is bound to sustain the combined effect of high thermal and inertia loads. The pressure coming on the crown is 980×10^4-$1\,470 \times \times 10^4$ N/m^2 (100-150 kgf/cm^2) and the crown temperature may vary within 623-873 K (350-600°C) depending on the cooling effect, material, and construction features. Particularly high are the thermal stresses set up next the combustion chamber.

The temperature of the piston is anything but uniform in operation—the crown is hotter than the skirt. The thermal expansion of the skirt along the axis of the piston pin is therefore greater than the swelling in the plane of connecting rod motion, resulting in the ellipticity of the skirt. To ensure an appropriate liner-to-piston clearance, the recourse is made to a taper crown or one of a diameter which is less than that of the skirt and also to recesses provided next the piston pin bosses.

Inadequate cooling of the piston may trigger cracks in the crown if the engine is overloaded. An overheating of the piston reduces the air charge drawn into the cylinder, leading to a loss of power. The problem is taken care of by cooling the piston with oil or water. In water cooling, the amount of heat removed will be much higher owing to the fact that the heat capacity of water is approximately 2.5 times that of oil. Therefore, water cooling is used on many diesel engines including the Sulzer RD-76 and RD-90 models and the MAN KZ 70/120 C model. The spilling of oil, if this is used as a coolant, may lead to the crown carboning. This will interfere with the heat transfer and cause overheating of the crown. On the other hand, the use of water as a coolant poses the problem of providing an adequate sealing which will prevent water from intermixing with the crankcase oil.

In use are one-piece pistons and built-up pistons. The crown of the pistons used on some crosshead diesel engines is made separately from the skirt.

Pistons are made from wear-resistant materials displaying satisfactory performance at elevated temperatures. Suitable in this case are cast irons with a tensile strength of 24-50 kg/mm^2 and a bending strength of 44-56 kg/mm^2 alloyed with chromium and nickel to improve their heat- and wear-resistant characteristics. The addition of nickel gives rise to a close-grained structure of the material.

Cast-iron pistons are rather heavy and find therefore application on low- and medium-speed engines. The crowns of built-up pistons exposed to high mechanical and thermal stresses are made from

medium carbon or low-alloy steel. To facilitate the running in, the skirt may be given a 0.010- to 0.015-mm coat of tin, on large bore engines, the alternative is to caulk in a wide ring in leaded bronze.

Pistons with cast steel crowns are widely used on low-speed high-output diesel engines made by Burmeister & Wain. The same applies to the Fiat 900S low-speed engine and the Pielstick PC2-5 medium-speed diesel. To reduce the pistons weight, and consequently the inertia loads, the Pielstick engines use steel crowns in conjunction with aluminium alloy skirts.

High-speed engines employ pistons made from aluminium alloys which are characterized by low density and high thermal conductivity. However, since the line expansion coefficient of aluminium alloy is 2-2.5 times that of cast iron, an appropriate cylinder clearance must be provided in engines with aluminium alloy pistons. An adequate warming up of the engine in this case is essential, otherwise compression blow-bys and the hammering of pistons against cylinder walls are likely to occur.

44. Piston Rings

Piston rings are classed as compression and oil-scraper rings.

Being fitted into grooves provided in the piston crown, compression rings serve to prevent blow-bys and convey the bulk of heat

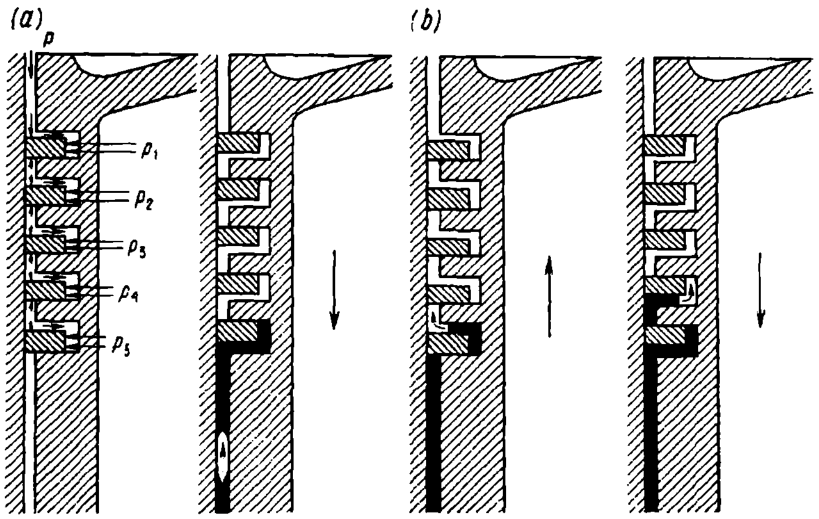


Fig. 38. Compression ring action

from the piston to the cylinder liner. The usual practice is to use between three and six compression rings per piston, high-speed diesel

engines use less rings than low-speed ones because of the fact that higher piston speeds lessen the possibility of a blow-by.

The sealing effect of compression rings is obtained partly due to the pressure of the gases entering the so-called back clearance between the piston and ring, and partly due to ring tension (Fig. 38*a*). A maximum gas pressure comes on the first and second rings next the combustion chamber. Further rings in the downward direction sustain a pressure which progressively decreases due to the throttling effect of the labyrinth formed by the back clearances. Being not over 9.8×10^4 N/m^2 (1 kgf/cm^2) per ring, the ring tension alone cannot ensure adequate sealing.

Compression rings operate under the conditions of boundary lubrication friction and elevated temperatures. To ensure ring tension and allow for the expansion of the rings as they become hot, joints are provided in the rings. Consequently, the ring diameter in a free state is always larger than the liner bore.

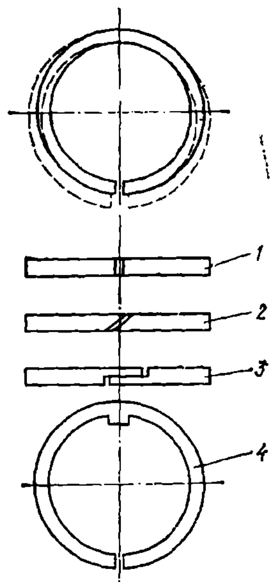


Fig. 39. Piston ring joints of various type

Referring to Fig. 39, piston ring joints can be of the straight *1*, bevelled *2* and stepped *3* types. Low-speed diesel engines employ piston rings with bevelled and stepped joints, high-speed engines resort to straight joints which are the most simple ones in manufacture. The fleeting operating cycle of high-speed engines does not permit the gases to pass through the straight joints of the piston rings. The compressed gap is 0.78-1.2 mm on the Ч 15/18 diesel, 2.0-3.0 mm on the Д and ДР 30/50 diesels and 2.0-2.5 mm on the R6DV-148 diesel engine.

Possible gas leakage is minimized by arranging the piston rings so that the joints should face different directions. However, the original arrangement of the piston rings fails to remain unchanged in service, for there are experimental proofs that the piston rings rotate in operation. On the one hand, this phenomenon appears to be of value, adding to the dependability of the piston rings. On the other hand, it is desirable that on the two-stroke engines the piston rings retain their original position unchanged so that no joint would appear against the ports in the cylinder liner and become trapped and broken when the engine is running. Special retainers or projections *4* made integrally with the piston ring serve to secure this in place. This practice is adopted on the 16ДРН 23/2 × 30 diesel engine.

The piston rings' thrust against the liner walls produces significant frictional forces influencing liner wear and the mechanical efficiency of the engine. The unit work of friction of a compression ring in a reciprocating piston is many times that of the cylinder liner. This implies that compression rings wear down more rapidly than liners do in spite of the fact that the material of compression rings is harder than that of cylinder liners by 20-25 BHN.

To facilitate the running in of piston rings and heat conduction from the piston, leaded bronze strips are press-fitted into the rings.

To improve wear resistance of piston rings, they are given a 0.10- to 0.15-mm coat of porous chromium. A more recent technique conducive to better wear resistance and longer service life is the coating of piston rings with molybdenum.

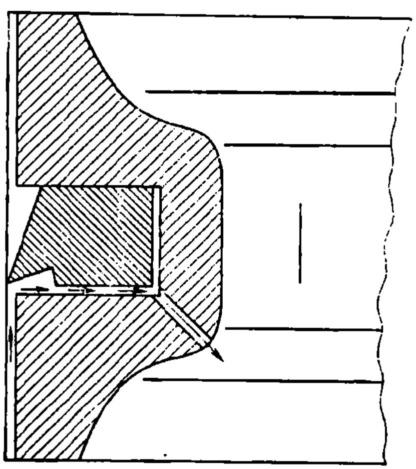


Fig. 40. Oil-scraper ring action

The material of piston rings is grey cast iron with a tensile strength of 21-24 kg/mm^2 and a bending strength of 40-44 kg/mm^2 which contains between 0.5 and 0.8% of phosphorus for better wear resistance. Sometimes use is made of alloyed cast iron with up to 1% of nickel and 0.8% of chromium that provides for low friction and adequate strength. On some high-speed diesels, the top-most piston ring is made from special steel.

The pumping action of compression rings is a phenomenon observed on trunk-type engines when their liner bore becomes overlubricated. Being responsible for the ingress of lubricating oil into the combustion chamber, it occurs on the following lines (Fig. 38*b*). When the piston is on the downstroke, the piston rings are pressed against the upper sides of the grooves, and oil enters the spaces below the rings. During the compression stroke when the piston travels upwards, the rings bear upon the lower sides of the grooves, and the oil is forced through the back and upper side clearances towards the combustion chamber.

To eliminate the possible ingress of oil into the combustion chamber, the piston skirt is fitted with oil-scraper rings. They are distinguished from compression rings by a sharp edge scraping off oil from the liner surface (Fig. 40). The oil-scraper rings in use are of various cross sections, e.g. plain with a 2-3 deg taper, keystone rings with a slot

at the middle, compound rings made up of two rings working in the same groove. The oil scraped off by rings edges during the downstroke is returned to the crankcase via oil drains provided in the piston. While on the upstroke, the ring bevelled side surfaces slide over the oil film without dragging this upwards.

45. Connecting Rods

The connecting rod is the most heavily stressed engine component bound to sustain during the compression stroke the combined effect of the stresses in bending and compression. To cope with the task of transmitting the gas pressure force to the crankshaft, without its longitudinal bending, the connecting rod must possess adequate strength and stiffness.

In trunk-type engines, the connecting rod links the crankpin directly to the piston. In crosshead engines, the piston rod and crosshead are interposed between the piston and connecting rod.

In Vee-type engines, the pistons of those cylinders, which are located in the same plane at right angles to the crankshaft axis, deliver power to the same crankpin. In this case, the connecting rod coupled directly to the crankpin is referred to as master connecting rod, and its counterpart linked to the crankpin by way of a knuckle pin is termed articulated connecting rod.

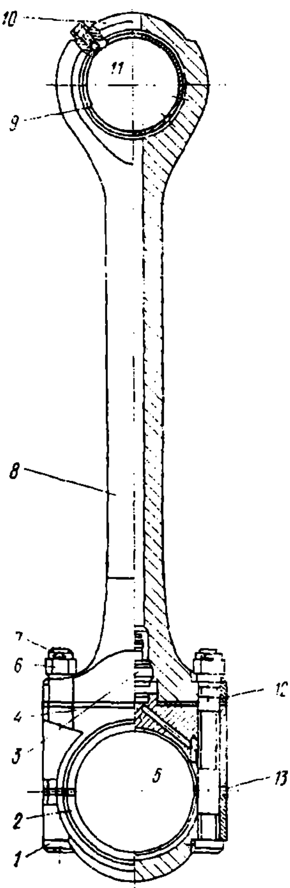


Fig. 41. Connecting rod

In the ordinary in-line engine (Fig. 41), the connecting rod consists of shank *8*, small end *10* and big end *4*. In trunk-type engines, the small end is made inseparable from the shank and is fitted with piston pin bearing *9* in the form of a bronze or steel bushing. Steel piston pin bearings are commonly lined with an alloyed bearing metal, such as babbitt (83 Sn). The material of non-ferrous bearings is deoxidized tin bronze. A set screw *11* prevents the bushing from rotation. Longitudinal or helical oil grooves may be provided inside the bushing to make for the piston pin lubrication.

Some diesel engines dispense with small ends at their connecting rods, the upper end of the shank is connected directly to the piston

pin by four studs in this case. This arrangement is met with on the 8ДР 43/61 diesel engine.

Crosshead diesel engines feature connecting rods with split small and big ends. Serving to couple the connecting rod to the crankshaft, a two-piece big end facilitates the engine erection and repair. This arrangement is particularly valuable in those marine main propulsion engines where the pistons are withdrawn upwards integrally with the connecting rod on removing the bearing bolts. The shorter the connecting rod, the easier the job can be coped with in the cramped engine room. A locating boss of the upper part of the big end serves to align the big end with the shank. The big end is fitted with steel shells *2* lined with a bearing metal (babbitt, leaded bronze). On rare occasions, the bearing metal is cast on the inside of the big end directly. The big end is connected to the shank by four bearing bolts *1* with castellated nuts *6* and cotter pins *7*. Assorted shims *13* are placed between the upper and lower parts of the big end to adjust the bearing clearance depending on wear. No shims are used on high-speed diesel engines with leaded bronze-lined bearings.

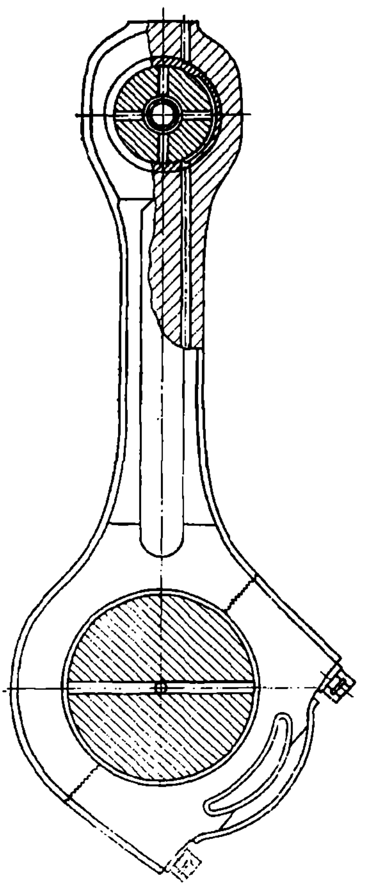


Fig. 42. Connecting rod with obliquely split big end

A shim *12* placed between the shank and big end controls the compression chamber height and, consequently, the engine's compression ratio. Depending on the engine speed the shank may be of the round, tubular or I-section. So, high-output low-speed diesel engines have connecting rods with the shanks of the round section to save on the cost of manufacture. In high-speed engines, the requirement of adequate strength and stiffness at minimum weight is met by connecting rod shanks of I-section. The material used for the manufacture of connecting rods is medium-carbon steel (0.35-0.45 C) or alloy steel (0.4 C, 1.5 or less Cr and Ni) or improved alloy steel.

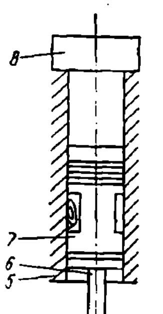
A supply of oil to lubricate the piston pin bearing and cool down the piston is fed over a central drilling *5* in the shank and check valve *3*. The piston pin bearings of high-speed low-output engines are splash lubricated by the cranks steadily dipping into the crankcase oil bath. Sometimes, an oil line fitted to the shank is used to lubricate the piston pin bearing.

The big end must be of a size enabling the withdrawal of the piston and connecting rod upwards through the cylinder. A plan followed on some medium-speed diesel engines in order to ease the job is the use of obliquely-split big ends (Fig. 42). In this case, it is possible to reduce the width of the big end, lessen the load on the bearing bolts and increase the diameter of the crankpin.

46. The Crankshaft

The crankshaft is a crucial and heavily stressed engine component. It is also a costly part, sometimes accounting for 20-25% of the cost of the engine. The crankshaft is bound to sustain fluctuating loads due to the gas pressure and the inertia forces of the unbalanced translating and gyrating masses which give rise to bending moments and torques of considerable magnitude.

Referring to Fig. 43, the crankshaft of a main engine comprises a plurality of crankpins and journals interconnected by means of webs. A flange at the aft end links the crankshaft to either a thrust



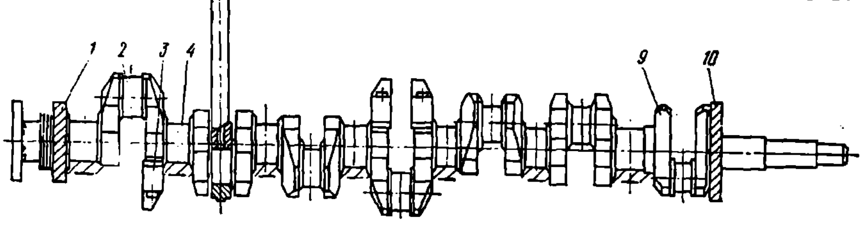


Fig. 43. Crankshaft and its location with reference to engine cylinder and piston

1—oil pump drive gear; *2*—crankpin; *3*—dynamic balance weight; *4*—main journal; *5*—cylinder liner; *6*—connecting rod; *7*—piston; *8*—cylinder head; *9*—scavenging pump-driving crank; *10*—camshaft timing gear

block shaft or an intermediary shaft of the ship's shafting. A gear fitted to the fore end provides drive for an oil pump and a water pump serving the needs of the engine. A camshaft drive gear is attached to the aft end of the crankshaft.

Taking the forces transmitted from the pistons to the crank-pins by way of the connecting rods, the big ends of which embrace the crankpins, the crankshaft rotates receiving support from the main bearings through the intermediary of its journals. The angular arrangement of the cranks relative to each other (crank angle) depends on the number of strokes and the number of cylinders.

For the balancing of the inertia forces of the gyrating masses and the moments these forces set up, dynamic balance weights are fitted to webs. In high-speed diesels, the balance weights are made integrally with the webs; in other engines, the weights *2* are secured to the webs *1* with the aid of recesses *4* and bolts *3* (Fig. 44).

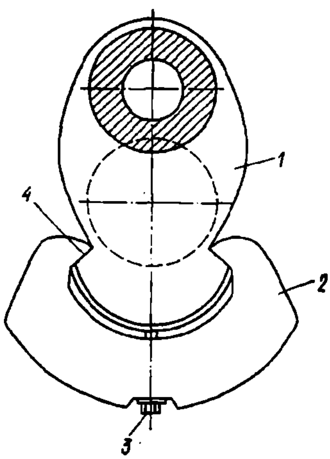


Fig. 44. Balance weights fitted to crankwebs

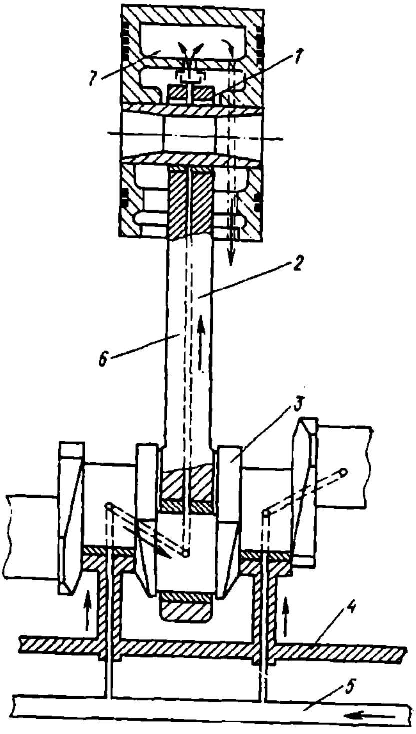


Fig. 45. Schematic of oil passages used to lubricate connecting rod-crank mechanism

To minimize the mass and, consequently, the inertia forces of the gyrating masses, the journals and crankpins are bored hollow. Blanking-off plates held fast by bolts or studs are provided at the end faces of the drilled journals and crankpins.

The number of cranks depends on the number of cylinders. Low-speed high-output engines employ crankshafts made up of two or three sections interconnected by flanges, posing thus less manufacturing problems than a one-piece crankshaft. For example, the crank-

shaft of the ДКРН 74/160 diesel engines having more than five cylinders consists of two sections.

The fabrication of crankshafts from individual journals, crankpins and webs is a cost-saving practice adopted by some builders of low-speed crosshead engines. Likewise in use are crankshafts built-up from one-piece forgings each consisting of two webs with the crankpin press-fitted therebetween as this is the case in some Sulzer engines and the Götaverken DM 850/1700 engine.

The crankshaft lubrication is taken care of as shown in Fig. 45. The main bearing shells *3* are lubricated by the oil pressure-fed from main oil gallery *5* over oil lines *4*. Through drillings in the webs, the oil reaches the crankpins and is forced through a drilling *2* in the connecting rod shank *6* towards the small end to lubricate the piston pin bearing *1* and to cool down the piston *7*.

In crosshead engines, the lubrication of the crankpin bearings is taken care of by the oil fed from the crosshead over the connecting rod drillings. The main bearings are lubricated by the oil from the main oil gallery. The oil pressure in the main and crankpin bearings is maintained within 39.2×10^4-$49 \times \times 10^4$ N/m^2 (4-5 kgf/cm^2), reaching 88×10^4 N/m^2 (9 kgf/cm^2) on high-speed units.

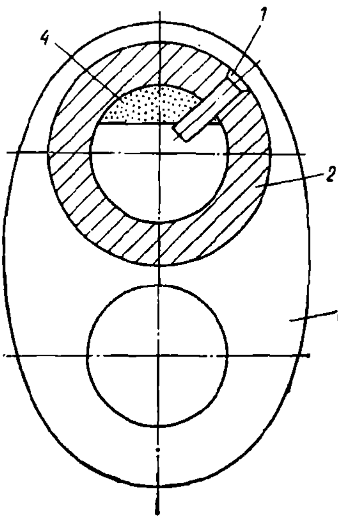


Fig. 46. Crankpin with centrifugal dirt excluder

1—dirt-excluding pipe; *2*—hollow crankpin; *3*—crankweb; *4*—sediments

The crankshafts of some high-speed diesel engines are provided with centrifugal dirt excluders (Fig. 46) inside their crankpins. A dirt excluder consists of pipe *1* expanded into the crankpin hollow and functioning as a centrifuge removing sediments from the oil. When the crankshaft rotates, the dirt and minute metal particles are thrown by centrifugal force towards the crankpin inside wall, and purified oil reaches the crankpin bearing via the pipe *1*.

The crankshafts of low-speed engines are commonly made from carbon steel of various grades (0.35-0.45 C), the material of high-speed crankshafts is steel alloyed with tungsten, nickel, chromium, magnesium (such as steels: 0.40 C + to 1.5 Cr, 0.40 C + to 1.5 Cr + + to 1.5 Mg; 0.18 C + to 1.5 Cr + to 1.5 Ni + to 1.5 W).

The forces due to the gas pressure and the inertia forces coming on the crankshaft journals and crankpins cause their wear after

a period in service. Consequently, these parts become elliptical and taper, and need reconditioning. To prevent this and render the journals and crankpins wear resistant, their surfaces are nitrided or given some other heat treatment. Peen hardening is also practised.

In low-speed diesel engines, the wear of crankshafts is relatively small. So, for example, the Götaverken main engine of the motor

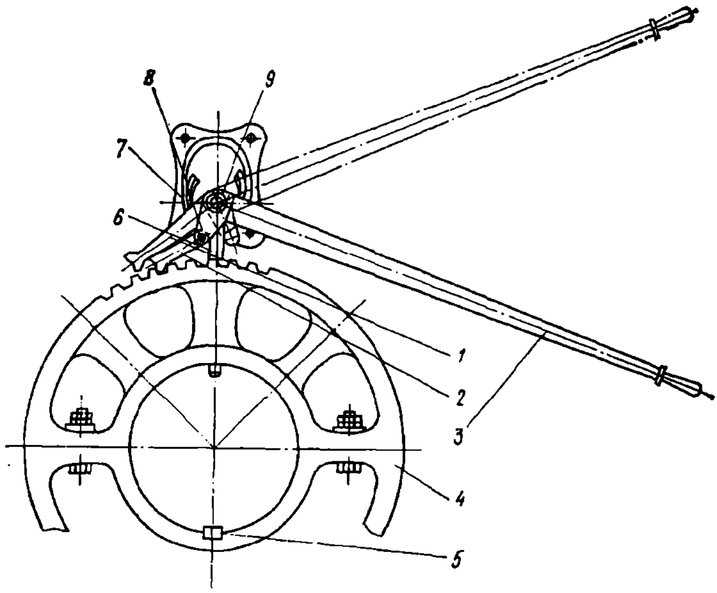


Fig. 47. Hand-operated jacking gear

1, 2—levers; 3—hand grip; 4—jacking gear wheel; 5—key; 6, 7—brackets; 8, 9—fulcrum pins

ship "Dmitriy Donskoy" was discarded on completing 130 000 running hours in 30 years of service when its 450-mm dia crankshaft showed a wear of 0.15 mm on the journals and 0.20 mm on the crankpins.

Engine jack-over is an operation frequently resorted to when adjusting the engine, circulating oil through the lubricating system, or in connection with the repair or renewal of piston and cylinder components. It is accomplished with the aid of a hand-operated, pneumatic or electric jacking gear, also termed barring or turning gear.

A hand-operated jacking gear is a very simple device consisting of two levers having different lengths which can be pivoted about a pin of a bracket secured to the engine structure (Fig. 47). In the

automatically-controlled main engine, the hand-operated jacking gear is replaced by pneumatic- or electric-driven mechanisms. To prevent the starting of the engine while the jacking gear is being engaged, the starting and reversing gear must be provided with a device to disconnect the starting air system from the air containers and to initiate alarm at the remote control panel.

47. Cylinder and Piston Malfunctions

Abundant piston and cylinder liner lubrication frequently causes the gumming of compression rings that leads to the thermal overstressing of these components, compression blow-by and loss of power. The high thermal stresses set up in the piston may trigger through cracks in the crown, while the gases ingress into the crankcase through the cracks may sometimes cause disastrous crankcase explosions.

The seizing of a piston is a trouble which may have a number of causes, such as inadequate lubrication giving rise to dry friction, extraneous articles in the cylinder (e.g. piston ring fragments or chipped off fragments of exhaust edges and scavenging ports), poor cooling of the piston and cylinder liner, lasting engine overload, poor fuel atomization, low standards of workmanship in erecting the engine (e.g. incorrectly set cylinder clearance, defective press-fitting of the piston pin leading to piston deformation), impaired oil lubricating capacity resulting from fuel leaks into the crankcase, etc.

For trouble-free operation of the pistons and cylinders it is essential to observe high standards of workmanship in erecting the engine. Effective cooling and lubrication are other requirements calling for attention. Engine overloads should be carefully avoided, and the fuel system should be always kept in good repair.

REVIEW QUESTIONS

1. What purpose does the connecting rod-crank mechanism serve in a diesel engine?
2. What are the main parts comprising the connecting rod-crank mechanism?
3. What purpose does the crosshead serve?
4. What are the main parts of a piston?
5. Why are pistons fitted with compression rings and oil-scraper rings?
6. How is the crankshaft linked to a piston?
7. What are the parts a connecting rod is composed of?
8. What forces act on the crankpins?
9. How is the lubrication of the crankshaft taken care of?
10. What are the main causes of cylinder and piston malfunctions?

Chapter VIII
THE VALVE GEAR

48. Valve Gear Function and Operation

The purpose of the valve gear is to ensure normal conditions for fuel combustion. In other words, the valve gear controls the clearing of the cylinders from the burnt gases and their filling with fresh air charges. In the four-stroke diesel engines, this is accomplished by a timed opening and closing of the inlet and exhaust valves, while in the two-stroke ones, by uncovering and covering the scavenging and exhaust ports.

In two-stroke diesel engines, this operation is carried out by loop scavenging (e.g. the Д and ДР 30/50 diesels), with the piston uncovering and covering the ports by the skirt in the course of the reciprocating motion. The two-stroke engines in which the scavenging ports are located higher than the exhaust ones employ automatic check valves to prevent the ingress of gases into the scavenging air receiver. In the engines with the scavenging ports located below the exhaust ones, as this is the case in the Sulzer RD diesels, rotary exhaust valves—provided in the exhaust manifold—close the exhaust ports when the scavenging ports are covered by the piston, and preventing scavenging air losses.

As it can be noted, the above arrangement dispenses with a specific valve gear which simplifies the engine design. However, the two-stroke diesel with uniflow scavenging features an exhaust valve in each cylinder head which is positively opened and closed by a valve gear. Scavenging air is admitted in these engines through the ports being uncovered and covered with the piston's skirt.

In the four-stroke diesel engines, a valve gear is indispensable. Referring to Fig. 48, the valve gear consists of inlet valve *5* and exhaust valve *9*, camshaft with cams *1*, tappet *3* with roller *2*, push rod *4* and rocker arm *6*. The valves are fitted into the cylinder head *8* and tightly pressed against their seats by springs. The camshaft is actuated by a gear train or a chain drive. Each of the valves is forced to open into the cylinder by a corresponding cam which lifts the tappet integrally with the roller against the action of the spring, causing the rocker arm to depress the valve spindle. The closing of the valves is effected by the springs.

The periods during which the valves stay open depend on the rate of crankshaft rotation. An optimum valve timing is determined experimentally by the builder. A correct valve timing is essential for the clearing of the burnt gases and the filling of the cylinders with fresh air charges.

When in their seats, the valves are tightly pressed against them not only by the springs, but also by the gas pressure in the cylinder. In high-speed marine engines, the valves are fitted directly into the cylinder head, in high-output diesels use is made of caged valves.

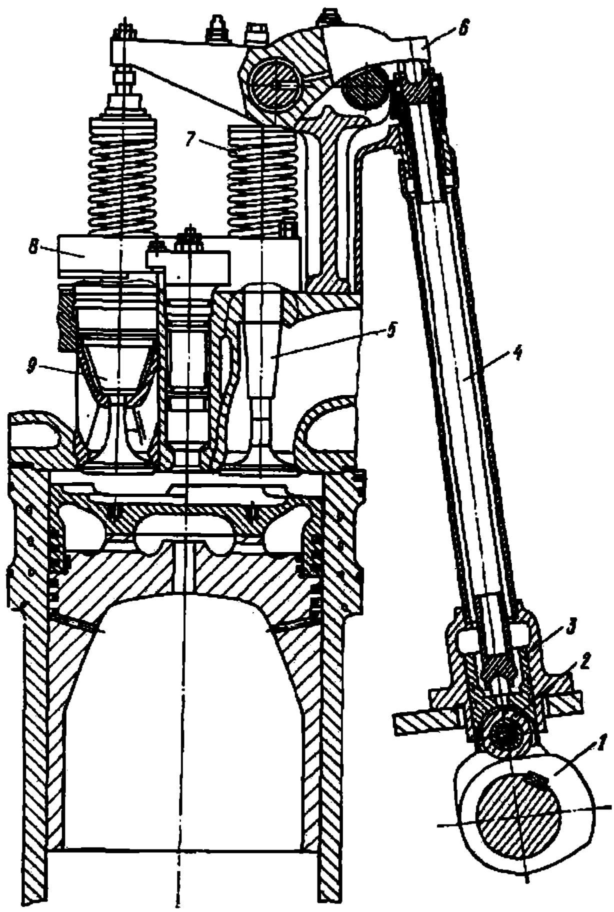


Fig. 48. Valve gear of four-stroke diesel engine

This practice is conducive to savings in operating cost, because a defective valve seat calls for renewing the cage only. The use of cageless valves necessitates the replacement of the entire cylinder head. However, cageless valves have a greater flow section, promoting a speedy clearing of the burnt gases and filling of the cylinder with fresh air. Yet, the flow section of valves cannot be increased

beyond a limit consistent with the size and strength of the cylinder head. The alternative resorted to on some high-speed engines is the use of two inlet and two exhaust valves, which not only enlarges the available flow section, but also lessens the friction losses in the ducting.

In operation, the valves sustain the combined effect of high live loads and temperatures heating up, for example the disc of the exhaust valve to 1 023-1 073 K (750-800°C) and that of the inlet valve to

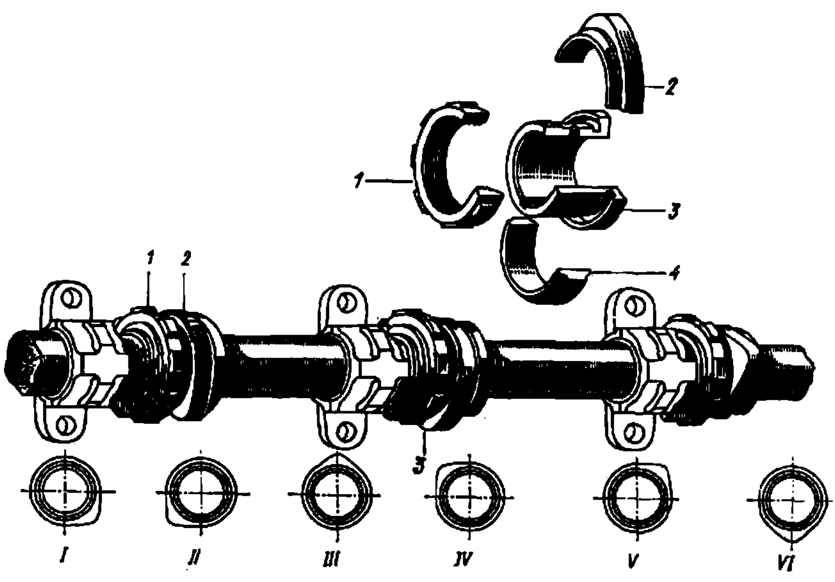


Fig. 49. Camshaft

1—clamping nut; *2*—upper half of cam; *3*—sleeve; *4*—lower half of cam; *I-VI*—cylinders

673-723 K (400-450°C). They are also exposed to the corrosive attack of the gases in the combustion chamber, which causes valve pitting.

The thermal expansion of the valve spindle and push rod is taken care of by setting an appropriate clearance between the striker bolt of the rocker arm and the cap at the valve spindle. The clearance is adjusted by turning the striker bolt up or down. A narrow clearance will prevent the valve disc from tightly fitting the seat so that a compression blow-by and the scorching of the disc will be imminent. On the other hand, a wide clearance will upset the valve timing and cause late opening and advanced closing of the valve. Moreover, a peening of the valve seat and disc is likely to occur in this case due to their impacts.

A point to be noted is that correct adjustment of the valve gear is essential for trouble-free and economical operation of the engine without output fluctuations and thermal overstressing of engine components.

It is obvious that the valves must be made from a material displaying high strength at elevated temperatures, resistance to gas corrosion and wear. Meeting these requirements are high-temperature chromium-nickel and silchrome steels (such as electric steels to 0.1 C, to 1.5 Cr, 8 Mn; 0.3 C, to 1.5 Cr, 8 Mn improved; to 1.5 Cr, 18 Ni, 0.25 C, etc.) which are commonly the material of exhaust valves. Inlet valves are made from low-carbon or low-alloy steels (such as steels 0.2 C; 0.45 C; 0.4 C, to 1.5 Cr; 0.4 C to 1.5 Cr and Ni, etc.). Valve facing with cobalt, chromium or stellite is widely used to improve the wear-resistant properties of valves, the coat applied being between 0.8 and 1.5 mm thick.

At the heart of the valve gear is the camshaft (Fig. 49) which is commonly located at about the mid-liner length, as this is the case for the 6ЧН 25/34 diesel engine, or lower. In high-speed Vee-type engines the camshaft is located within the Vee, serving both banks of cylinder heads (e.g. the 16ЧН 26/26 diesel). Also met with in high-speed engines is an overhead camshaft located on the cylinder head above the valves. In this latter case, the valves are operated by the cams directly or through the rocker arms fitted to the cylinder head. Neither push rods nor tappets are required with overhead camshafts which is an asset of this configuration conducive to a reduction of inertia loads so important for high-speed units. However, the mid-liner length or lower arrangement of the camshaft appears to be more convenient as far as the engine's erection and operation are concerned, and has come, therefore, into wide-spread application in marine practice.

Camshafts are either single-piece parts or they are made up of several sections joined by flanges. The drive is effected by crankshaft *4* through the intermediary of a gear train in most cases (Fig. 50) comprising timing gears *3* and *1* fitted to the crankshaft and camshaft, respectively, interconnected by means of idle gear *2* (some camshaft drives feature more than one idle gear). In the engines where the crankshaft and camshaft are located a wide distance apart, as this is the case on high-output low-speed diesels (e.g. the 8ДКРН 74/160 type), the camshaft is actuated by a chain drive (Fig. 51).

The material of camshafts is low-carbon or alloy steels. The cams serving to operate the valves are either fitted to the camshaft or are made integrally therewith, this latter practice is adopted predominantly on high-speed small engines (such as 0.15 C, to 1.5 Cr improved; 0.12 C, to 1.5 Cr & Ni etc.). Some low-speed diesel engines feature built-up cams, each consisting of two halves. Referring to

Fig. 49, which depicts the camshaft used in the Д and ДР 30/50 diesel engines, it can be seen that each cam consists of two halves held fast to each other by a coupling nut when this is tightened home on

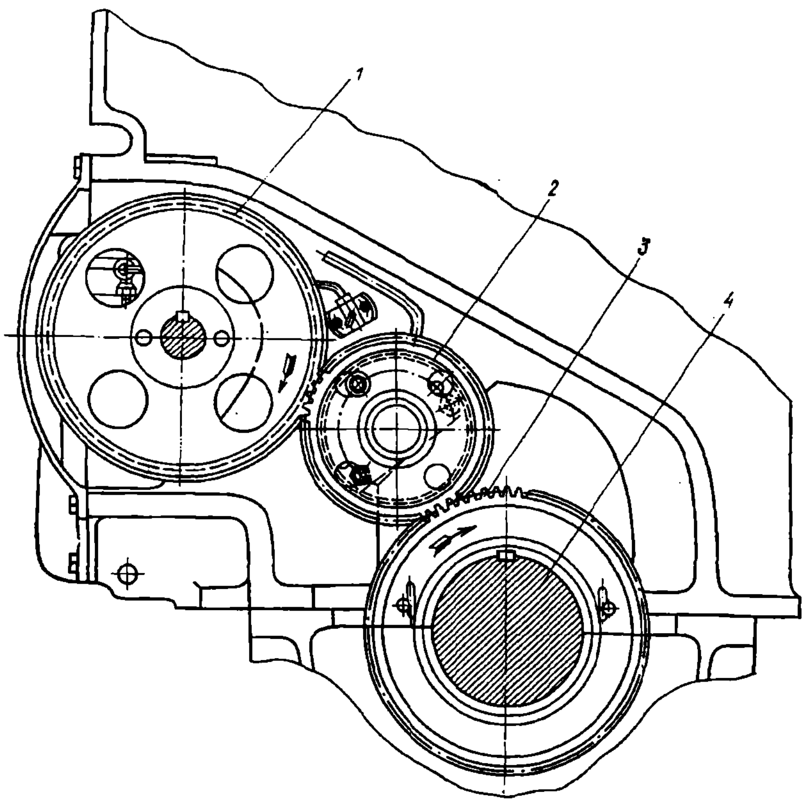


Fig. 50. Camshaft timing gear train

a sleeve keyed to the shaft. Built-up cams simplify the manufacture of camshafts, provide for a saving in maintenance cost and facilitate engine tuning up.

The cams are fitted to the shaft so that fuel injection and valve operation are effected strictly in accordance with the firing order. Their number varies with engine design. In a four-stroke direct-reversing engine each valve is operated by a pair of cams, each one being used depending on whether the ship is steaming ahead or astern. In the latter case, the camshaft is shifted axially so that the valve tappets are registered with the astern cams.

The shape of the cam determines the motion of the valve it operates, controlling the opening of the valve and the period during which it remains open. The flanks of the cams are case hardened and polished for good wear resistance. The material of split cams is low-carbon or low-alloy steel (such as steels having 0.15 C; 0.15 C to 1.5 Cr & Ni; 0.40 C to 1.5 Cr, etc.).

Trouble-free operation and good economy of an engine are influenced by the inertia forces of the valves and valve gear, the velocity of the incoming air charge and the exhaust gases, and the efficiency of valve cooling.

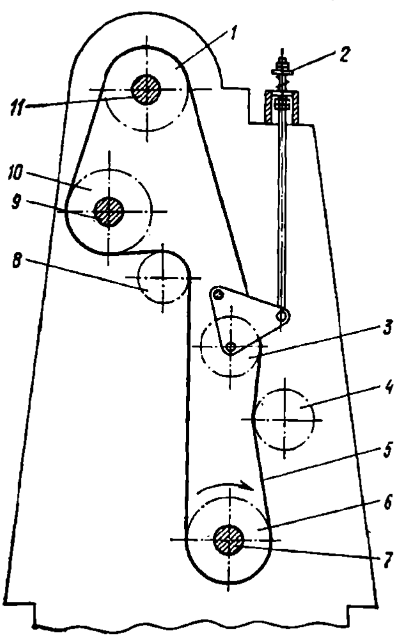


Fig. 51. Camshaft chain drive

1, 3, 4, 6, 8, 10—sprockets; *2*—chain tensioner; *5*—chain; *7*—crankshaft; *9*—camshaft; *11*—auxiliaries drive shaft

49. Exhaust Valve Rotators

In spite of being made from wear- and corrosion-resistant materials ensuring strength and gas tightness under the conditions of high pressure and temperature the exhaust valves are bound to operate, some diesel engines have the exhaust valves forcibly rotated in operation so as to prevent their non-uniform heating and burning.

The devices materializing this provision are termed exhaust valve rotators. Fig. 52 depicts a valve rotator known under the trade-name Rotocap which finds application in Pielstick engines, providing for the rotary motion of the exhaust valves at a rate of 3-4 rpm. Essentially it consists of static body *3* (also shown at *11*), disc spring *4* and valve spring retainer *7*. The body *3* is fitted to valve guide *2* and cylinder head *9*. It is provided with sectorized recesses arranged all the way along the circumference of disc spring *4* which, in its turn, is supported by a projection of body *3* so as to bear against spring retainer *7* with its opposite side. Accommodated in each recess is steel ball *8* with return spring *1*. The spring retainer *7* supports spring *5* of exhaust valve *10*.

As long as the exhaust valve is closed, the valve spring is tensioned but slightly so that the disc spring is not deflected. With the opening of the valve, the valve spring is compressed, exerting

an action on the disc spring through the retainer. The resulting deflection of the disc spring causes the steel balls to roll down the inclined guides towards a deeper part of the recesses, inducing the disc spring to turn integrally with the spring retainer and, consequently, the valve spring and the valve proper. Cotters 6 facilitate the rotary motion of the valve. When the valve starts closing, the valve spring expands, relieving the disc spring of the pressure. The disc spring resumes its original shape, freeing the balls which are returned by their springs into the original position. After that the cycle is repeated.

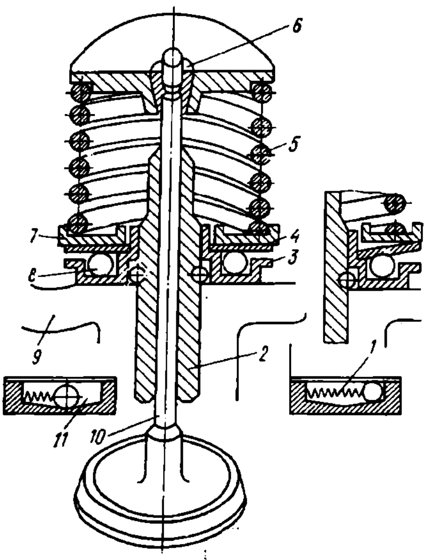


Fig. 52. Rotocap exhaust valve rotator

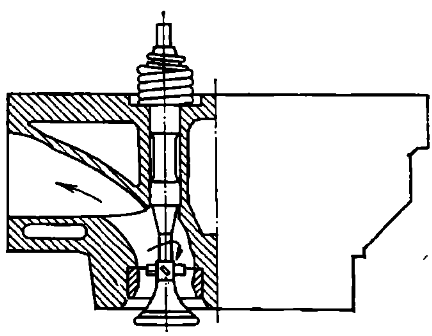


Fig. 53. Vane-type exhaust valve rotator

Depicted in Fig. 53 is an exhaust valve rotator used in MAN diesel engines. Claiming simplicity and trouble-free operation, this device takes the form of vanes provided on the valve spindle which cause the valve to rotate under the influence of the exhaust.

50. Scavenging and Exhaust Systems

The clearing of burnt gases and the charging with fresh air take place in a two-stroke diesel engine under worse conditions than this is the case in a four-stroke one. This is because the burned gases are removed by the scavenging air rather than by the piston, and the process is confined to a fraction of a crankshaft revolution only. So, the scavenging air mixes with the burned gases and the cylinders are deprived of a wholesome air charge.

Depending on the pattern of scavenging air flow, two-stroke diesel engines fall into loop-scavenged and uniflow-scavenged units depict-

ed in Fig. 54. In the former, the scavenging air enters the cylinder through the scavenging ports, first ascending to the cylinder head and then descending to the exhaust ports on turning at the top of its path as this is shown in Fig. 54*a* through *c*. In the latter, the scavenging air follows a straight path along the cylinder axis as this can be seen in (*d*) and (*e*). Controlling the pattern of flow are obviously the positions of the scavenging and exhaust ports relative to each other, the height of the ports and their attitude with respect to the cylinder axis.

Consider the design features of the main scavenging systems.

Referring to Fig. 54*a*, the cross loop port scavenging system is

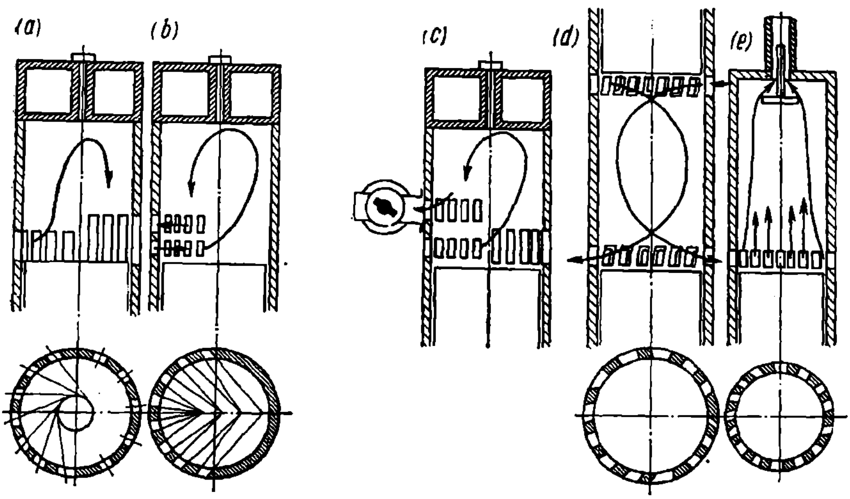


Fig. 54. Schematic of scavenging used on two-stroke diesel engines

characterized by the arrangement of the scavenging and exhaust ports at the opposite walls of the cylinder. If viewed in cross section in plan, the scavenging ports are located eccentrically and have the edges of longer sides bevelled in the vertical plane. The exhaust ports are longer than the scavenging ones. As a result, these ports are uncovered by the piston travelling towards BDC earlier than the scavenging ports, permitting a free escape of the burned gases before scavenging air is admitted into the cylinder from the receiver. Supplying a steady flow of scavenging air and posing no problems in manufacture and operation, this system is not free from shortcomings. Inherent in it is a loss of scavenging air through the exhaust ports which stay uncovered together with the scavenging ports when the piston is at BDC and also for a while after the scavenging ports

have been covered by the piston. Also an incomplete clearing of the burned gases takes place because some of them mix up with scavenging air, causing excessive fuel consumption. The cross loop scavenging is commonly used on diesel engines with an output of 73.6-110 kW (100-150 hp) per cylinder.

The loop port scavenging system (Fig. 54*b*) relies for operation on a two-tier arrangement of the exhaust ports and scavenging ports at one side of the cylinder, the former superposing the latter. The port edges are bevelled in the vertical plane to direct the air flow against the cylinder wall. The loop-type pattern of air flow provides for a satisfactory clearing of burned gases, as this has been proved experimentally, and is a feature of MAN diesel engines. However, some of the air charge may be lost through the exhaust ports which are uncovered after the piston travelling towards TDC has covered the scavenging ports. To prevent this, the MAN K7Z 70/120C diesel engine installed in the motor ship "Ugleuralsk" was fitted with rotary exhaust valves which closed the exhaust ports on admitting the air charge into the cylinders. However, frequent troubles developed by the rotary valves and their actuating mechanism have forced the builder to abandon the practice of using them. Recent MAN diesels have no rotary exhaust valves.

Loop port scavenging with controlled exhaust valves of the rotary type is used in the Sulzer 6RSAD-76 and 9RD-90 diesel engines (Fig. 54*c*). In the RD-90 engine, the liner is provided with main scavenging ports and a number of auxiliary scavenging ports which add to the flow section and promote efficient cooling of the piston at all sides, lessening the thermal stresses. The exhaust ports superpose the auxiliary scavenging ports. The scavenging air entering the cylinder flows around its walls and may be partly lost through the exhaust ports which stay uncovered for some time while the piston is travelling towards TDC. To prevent this, the exhaust duct is fitted with a rotary valve set to close the exhaust ports at the instant the piston travelling upwards covers the scavenging ports. However, the rotary valve and its actuating mechanism introduce complicacy into the diesel engine design.

The uniflow port scavenging system features an upper arrangement of the scavenging ports around liner circumference, and a lower arrangement of the exhaust ports. Since in this case the scavenging air follows the shortest path between the ports, a mixing of the air with the burned gases is practically excluded. Employed in opposed-piston diesel engines (Fig. 54*d*), this scavenging system calls for arranging the lower crankshaft so as it would rotate 8-12 deg in advance of the upper one. Consequently, the upper piston uncovers the scavenging ports later than the lower piston has uncovered the exhaust ports with the result that the pressure in the cylinder sharply decreases. This promotes the charging with extra fresh air which

enters the cylinder after the exhaust ports have been covered.

The uniflow valve scavenging system provides for admitting scavenging air by way of the ports located at the bottom of the cylinder liner around its circumference and for disposing of the burnt gases through one or more than one camshaft-actuated exhaust valve fitted in the cylinder head (Fig. 54*e*). The valve in conjunction with the valve gear lending itself to adjustment create the prospect of setting an optimum valve timing which would prevent any loss of the air charge. The strategically placed scavenging ports ensure an efficient cooling of the piston which reduces the thermal stresses therein. The uniflow valve scavenging system is employed in diesel engines made by the Bryansk Engineering Works in this country. It is also a feature of diesel engines made by Burmeister & Wain, Stork and other builders.

51. Valve Gear Malfunctions

Exhaust valves operating at elevated temperatures, which may distort a valve disc and provoke leakage, are a frequent cause of troubles. Some of the air charge admitted into a cylinder may escape into the exhaust manifold through a leaky valve during the compression stroke. This will reduce the compression pressure and temperature, inviting loss of power and difficult cold starts of the engine. The disc of a leaky valve may burn so as to render the valve inoperative. Leaky valves are reconditioned by turning the valve disc and seat in a lathe and grinding the valve to the seat.

Inadequate lubrication of the camshaft leads to rapid wear of cams and even to the scoring of cam flanks. Defective cams must be renewed, and the lubricating system must be checked for providing oil to all lubrication points.

If the camshaft is of the chain-driven arrangement, the chain tension must be adjusted at regular intervals, and the worn chain links must be renewed. If a gear train is used to drive the camshaft, the defective gears, i.e. those badly worn down, cracked, chipped or damaged in some other way, must be renewed.

REVIEW QUESTIONS

1. What purpose does valve gear serve?
2. What are the main components of valve gear?
3. Why is the rate of camshaft rotation only half that of crankshaft rotation in a four-stroke diesel engine?
4. What purpose does an exhaust valve rotator serve?
5. How is air introduced into the cylinders of two- and four-stroke diesel engines?
6. Discuss possible malfunctions of valve gear.

Chapter IX

THE FUEL SYSTEM

52. Main Components of the Fuel System

The fuel system caters for the delivery of fuel into the cylinders under any load on the engine. Since the mixture of fuel and air takes place in diesel engines inside the cylinder, the system for feeding and injecting the fuel must meet the following requirements:

— trouble-free operation throughout the period between planned preventive maintenances and inspections as specified in an appropriate maintenance manual;

— feeding fuel in a strictly specified amount in order to enable the engine to cope with a given load;

— the injection of fuel into the engine cylinders in an optimal prescribed way creating the most favourable conditions for combustion;

— ease of attendance in operation, including the replacement of fuel injectors and fuel pumps;

— stable operation at low engine speed;

— immediate readiness for failure-free starting and shutting down of the engine.

The fuel system must include means for bunkering, filtering and conditioning and feeding the fuel timely into the engine cylinders in amounts necessary to carry the engine load. Accordingly, the fuel system of a marine diesel engine comprises service tanks, fuel transfer and fuel booster pumps, primary and fine filters, preheaters and separators, high-pressure fuel filters, injectors, fuel-injection pumps, low- and high-pressure fuel lines. Referring to Fig. 55, fuel transfer pump *2* feeds fuel from storage tank *3* into service tank *7* located above the engine to produce a head and fitted with overflow pipe *6*, fuel content gauge *8* and drain cock *9*. The outflow from service tank *7* reaches fuel booster pump *11* through duplex prime filter *10*, being then delivered by this latter pump to high-pressure fuel pumps *13* via duplex fine filter *12*. The high-pressure pumps feed fuel to injectors *17* through high-pressure lines *15* and high-pressure edge-type filters *16*. Surplus fuel which has not been injected due to cutoff is removed from the fuel-injection pumps by return pipe *14*. Any fuel leakage from injectors or fuel-injection pumps is drained into collecting tank *19* or back into the service tank over leak-off line *18*. Separator *21* with preheater *22* provided between the storage tank and service tank is used to remove water and sediments from fuel. In the event of a pressure buildup downstream of duplex fine

fuel filter *12*, relief valve *20* diverts the fuel flow to the service tank. For bunkering use is made of port and starboard deck sockets *5* and lines *4*. At *1* is shown a stand-by fuel transfer pump. The storage tank is provided with stripping pump *23* used to drain it from sludge. If the main engine digests residual fuel, a service tank for diesel fuel used in starting and maneuvering the engine is included into the fuel system.

It is the usual marine practice to stow fuel in double-bottom tanks. Apart from that, an emergency fuel reserve sufficient for steaming

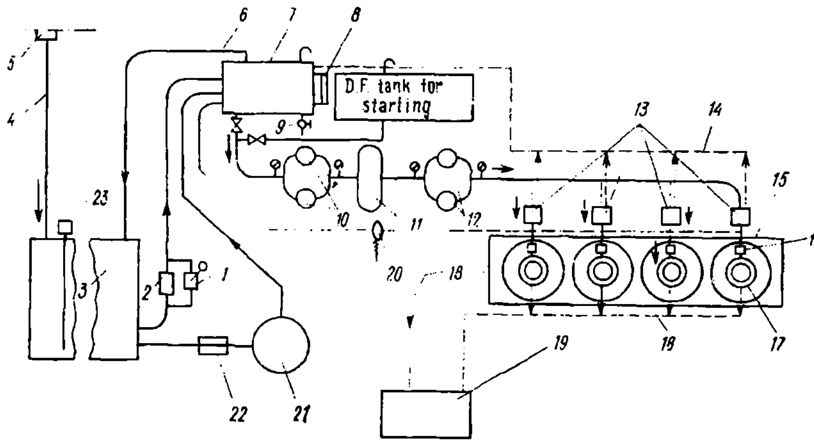


Fig. 55. Schematic of fuel system

during a period of at least 24 hours is provided outside the double bottom for use in case of bilging.

Each ship is to be fitted with service tanks for the main and auxiliary engines, a stand-by tank for the fuel treated in a separator, settling and collecting tanks. The tanks holding residual fuel must be provided with a thermal insulation.

As mentioned above, viscous residual fuel with a high loading of sediments requires careful filtration and centrifugation preceded by a heating in the tank.

Residual fuel must be also preheated to 338-368 K (65-95°C) before the fuel-injection pumps. Then, its temperature before the injectors will be 323-333 K (50-60°C), enabling normal atomization to take place. Since the fuel filters get clogged after a period in service, their performance is monitored by differential pressure gauges measuring the fuel line pressure before and after a filter.

53. Fuel Filters and Separators

Trouble-free and efficient performance of the fuel system is dependent chiefly on the thoroughness with which the sediments and other impurities are removed from fuel. The sediments present in fuel are hazardous for the fuel equipment, provoking its rapid wear. To prevent this, the fuel system incorporates strainers, primary and fine filters, high-pressure edge-type filters and separators. The multi-stage system of filtration safeguards the injector orifices

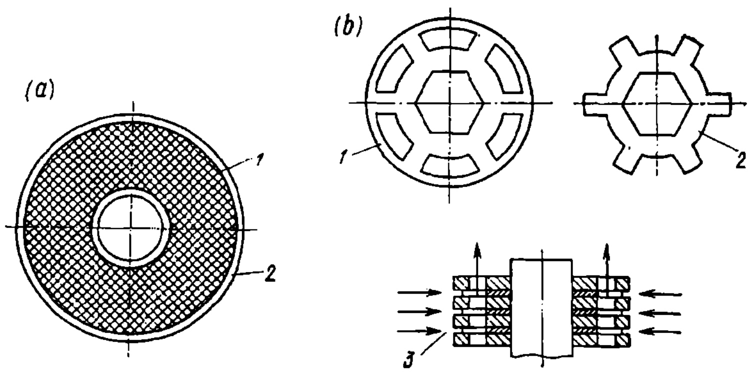


Fig. 56. Fine filter elements

against plugging, minimizes the possibility of scoring the plungers and barrels of the fuel-injection pumps as well as the nozzl bodies and valves of the injectors.

Strainers treat the fuel between the service tank and the fuel transfer pump, removing the coarser impurities. Prime filters provide protection to the components of the fuel system by removing the particles over 2 μm from the fuel. Fine fuel filters exist in a variety of types considered below.

So, depicted in Fig. 56*a* is an element of a fine-mesh filter in the form of brass wire gauze disc *1* with rim *2* and meshes between 0.1×0.1 and 0.16×0.16 mm. Another fine edge-type filter is illustrated in Fig. 56*b*, consisting of a stack of perforated discs *1* and spacers *2* which form passages *3* between 0.05 and 0.10 mm wide for the fuel to pass and the impurities to become settled in them. Any dirt or sediment accumulated between the discs is scraped off with the aid of thin metal fingers interposed between successive discs with one of their ends while the other ends are attached to a shaft with a handle at a top which can be turned by hand.

Referring to Fig. 57, a fine-mesh filter consists of housing *1* containing a stack of brass wire gauze discs *2* separated by spacers *7*

and assembled on hollow centre rod *8* drained through which is the filtered fuel. The housing is provided with inlet unions *5* and *6*, an air-bleed valve, three-way valve *4* and filtered fuel outlet *3*. The filter consists of two identical units used alternately, enabling one of the units to be taken apart and cleaned while the other unit

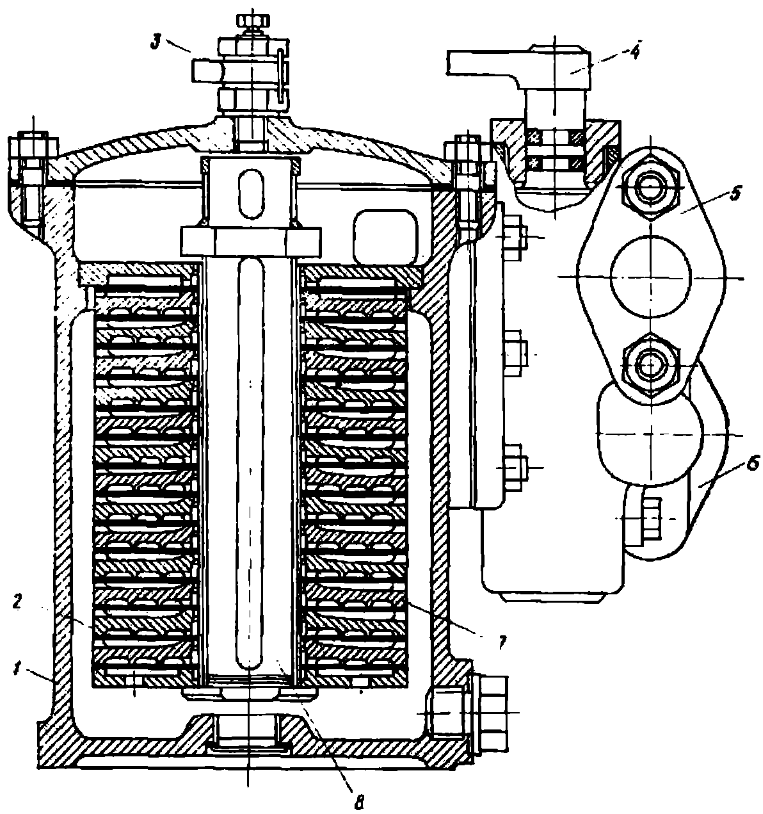


Fig. 57. Fine-mesh filter

is in operation. The changing-over from one unit to the other is effected by means of the three-way valve. The number of filtering discs in a stack must be a minimum one which offers a minimum resistance to the flow and provides for an adequate filter performance. When the filter is clogged, its resistance increases, being betrayed by the pressure differential of the filter which exceeds the normal 2 840-4 900 N/m^2 (0.3-0.5 kgf/cm^2) required for trouble-

free engine operation. The remedy in this case is urgent cleaning of the unit from the impurities.

Fine fuel filters come under the category of what may be called internal-surface filters, the impurities being trapped inside filtering elements made from cloth, cardboard, paper or felt. A felt filter (Fig. 58) consists of a stack of felt discs *1* and gauze *2*. A silk bag is slipped on the replaceable filtering element to prevent the ingress

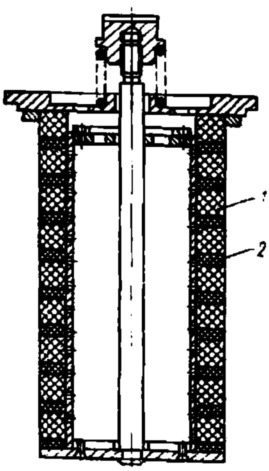


Fig. 58. Felt-type fine fuel filter

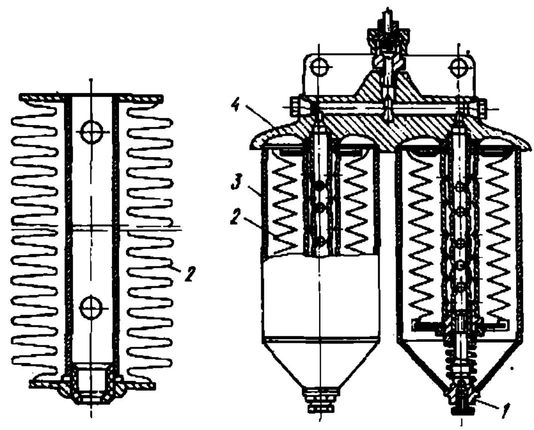


Fig. 59. Type 2TФ-4 fine fuel filter

of felt fibers into the fuel system. The felt filter is capable of removing particles ranging in size between 5 and 50 μm from the fuel. Novel filter media gaining ground nowadays are porous bronze and ceramics.

Diesel engines with an output over 220 kW (300 hp) made in this country employ standard fine fuel filters of the 2TФ-4 and 2TФ-5 type with a throughput of 200 and 1 200 kg/h, respectively, depicted in Fig. 59. Each unit consists of two filtering elements *2* contained in separate housings *3* superposed by common cover plate *4*. The fabric filtering element of the octahedral configuration offering a maximum filtering surface for a given volume is capable of removing sediments measuring 5 μm and upwards. After a period in operation, the filtering element is washed by reversing the flow of fuel through it with the aid of a three-way valve. The sediments removed by the fuel flow from the filtering element are disposed of through a drain plug *1* provided in the bottom of the housing. Clean fuel issuing from the drain hole indicates that further washing is not required.

The fuel system of the ships operated on residual fuel includes centrifugal separators of the disc or tubular type to remove sediments and water from the fuel.

54. Fuel Booster Pumps

To induce a continuous flow of fuel from the service tank to the fuel-injection pumps, the recourse is made to booster pumps incorporated in the main fuel line. A booster pump must deliver under a pressure sufficient for trouble-free functioning of the system, i.e. it must overcome the frictional resistance of the filters, lines and fitting with an allowance for an increase in filter resistance with time, prevent the bubbling of the fuel and create a positive suction

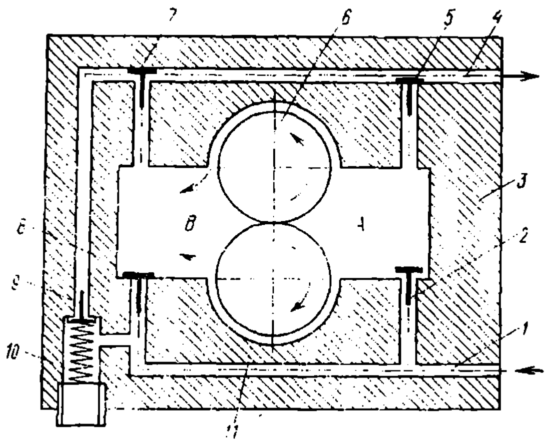


Fig. 60. Gear-type pump

lift at the inlets into the fuel-injection pumps. The booster pump delivery pressure commonly varies between 1.96×10^4 and $29.4 \times \times 10^4$ N/m^2 (0.2-3 kgf/cm^2).

Marine practice makes use of fuel booster pumps of the gear, reciprocating, vane and centrifugal type. Consider the design features and operating principles of these pumps.

Predominantly used in booster applications is the simple, compact and dependable gear pump capable of creating a suction lift of 1-1.5 m H_2O. It consists of body *3* (Fig. 60) with power impeller gear *6*, idler impeller gear *11* and two sets of suctions valves *2*, *8* and delivery valves *5*, *7*. The power impeller gear is actuated by the camshaft or governor shaft; in use are also crank drives. The two

sets of suction and delivery valves enable the pump to deliver fuel always in the same direction irrespective of the direction of crankshaft rotation. The fuel sucked from the service tank through passage *1* enters the suction side *A* through open valve *2* from which it is carried by the rotating impeller gears *6* and *11* over the clearance they make with pump body *3* into the delivery side *B*. By-pass valve *9* loaded by spring *10* provides for a constant pressure of the fuel flow. When the power impeller gear is rotating counterclockwise, suction valve *2* and delivery valve *7* are open whereas valves *5* and *8* are closed. When the rotation of the impeller gears is reversed, so that the suction side is at *B* and the delivery is at *A*, the fuel admitted into the pump over passage *1* enters the suction side through open valve *8* and is delivered through open valve *5*. Valves *2* and *7* are closed in this case.

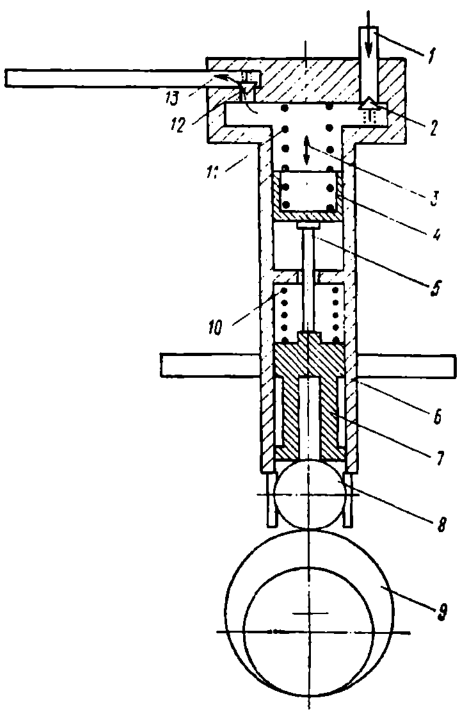


Fig. 61. Reciprocating fuel booster pump

Reciprocating booster pumps are commonly used in smaller diesel engines of the automotive type. Their design features and operating principle are depicted in Fig. 61. It can be seen that the pump consists of body *6* reciprocating in which is piston *4* and rod *5* coupled to roller *8* which contacts cam *9* setting the pump into motion. Suction valve *2* and delivery valve *12* are provided at the top of the pump body. Their operation is influenced by relevant springs *10* and *11*.

When the piston acted upon by the piston spring goes downwards and a vacuum is created above the piston, the delivery valve closes and the suction valve opens, admitting fuel from passage *1* into space *3*. At the same time, the fuel contained in the space below the piston is delivered into fuel line *13* over a by-pass passage. When the lobe of the cam *9* comes under the roller *8*, tappet *7* and rod *5* cause the piston to displace upwards against the action of piston spring *11*. The fuel in space *3* is delivered into line *13* over delivery valve *12* while suction valve *2* is held closed due to the pressure of

the fuel. Some of the fuel thus delivered is expelled over the by-pass passage into the space below the piston.

The reciprocating pump is capable of automatically changing its stroke and, consequently, the delivery depending on the delivery pressure. Should the pressure in line *13* exceed the force of piston spring *11*, the pressure existing below the piston causes it to lift clear of the rod which is not attached to the piston. Becoming thus disengaged from the tappet, the piston will not displace downwards and

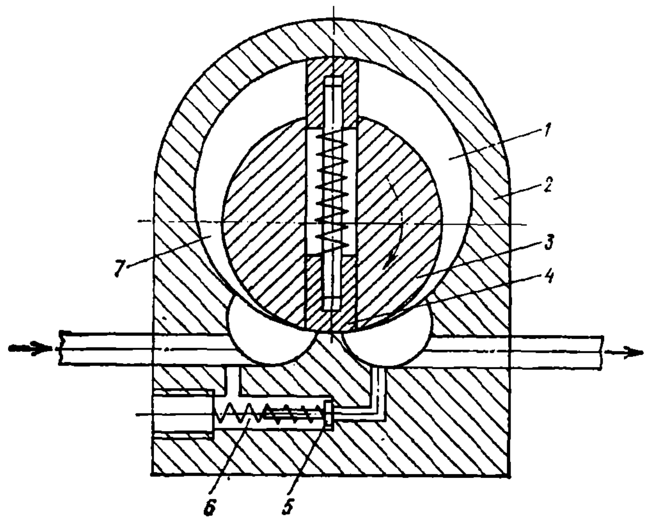


Fig. 62. Vane pump

the delivery will be interrupted. However, the pump will automatically resume operation as soon as the pressure in the delivery line falls below the pressure created by the piston spring and the piston becomes settled down on the rod.

A reversing of the crankshaft rotation does not change the direction of fuel delivery owing to tappet spring *10* keeping roller *8* in contact with the cam at all times irrespectively of whether the pump is in operation or idle. Reciprocating fuel booster pumps are in limited use because of the pressure fluctuations in the delivery line.

The vane pump (Fig. 62), being as compact as the gear type and capable of producing a significant suction lift and operating against an adequate delivery head, is in wide-spread use on small high-speed diesel engines of the unirotational type. The pump consists of case *2* eccentrically rotating in which is rotor *3* with slots accommodating spring-loaded vanes *4*. The rotor shaft is offset with respect to the axis of the cylindrical casing bore through a distance equal to the

eccentricity. The vanes subdivide the casing into suction side *1* and delivery side *7*, and a variable clearance, the rotor produces with the casing due to the eccentricity, provides the space through which the vanes displace the fuel from the suction to the delivery side. By-pass valve *5* loaded by spring *6* is fitted in the delivery side to provide for a constant delivery pressure. If the pressure in the fuel line exceeds the by-pass valve-opening pressure set with the aid of a screw, valve *5* by-passes some of the fuel into the suction line. In operation, a vacuum created behind the vanes due to the rotation of the rotor causes fuel to enter the suction line connected to the service tank. A fuel head set up at the same time before the revolving vanes provides for the functioning of the fuel system with a marging required to overcome the frictional resistance. The pump capacity varies with the engine speed. To minimize delivery fluctuations, the rotor is commonly fitted with several vanes.

55. High-Pressure Fuel-Injection Pumps

A high-pressure fuel-injection pump serves to feed fuel in a metered amount decided by the load at an appropriate instant of the operating cycle. Accordingly, it consists of a means of metering and a means of feeding which exist in a variety of types.

The fuel-injection pumps in use are of the cam-operated reciprocating type featuring a constant or variable stroke (in marine practice preference is given to the constant-stroke kind) with the throttling of the fuel flow during either the suction stroke or by-pass. The amount of fuel delivered per stroke is varied by means of by-pass valves or plunger-controlled ports in three different ways, i.e. by changing the point at which the delivery begins or ends or by changing both these points. In use are single or multiple pumps, the last-named variety consisting of a number of pump elements contained in a common housing.

The fuel-injection pumps with a changing point of the beginning of delivery employing the so-called control by timed by-pass (Fig. 63) find application mainly on low-speed diesel engines such as, for example, those of the Д and ДР 30/50 ranges and some of the Sulzer engines. They are not suited for high-speed installations because of a significant mass of the moving parts and a complicated suction valve gear. Apart from that, the pumps in question have an intricate control system and are likely to cause injector afterdripping resulting from a sluggish plunger performance towards the end of delivery at low engine speeds.

The means of feeding of the pump with control by timed by-pass shown at (*a*) consists of plunger *15* actuated by cam *11* of camshaft *12* through the intermediary of tappet *13* with roller *10* against the action of spring *14* returning the piston into its original position

after the stroke. The means of metering comprises suction valve *2*, lower tie-rod *7*, upper tie-rod *4* and cut-off lever *9* fitted to control eccentric *8*. Lower tie-rod *7* is provided with screw *6* and a locknut which serve to adjust tie-rods *7* and *4* for length. The suction valve admits fuel into space *1* above the plunger during the suction stroke and controls the amount of fuel delivered to the injectors, being actuated by tie-rod *7* resting on cut-off lever *9* which, in its turn, is actuated by tappet *13*. When the plunger is at TDC, a clearance *a*

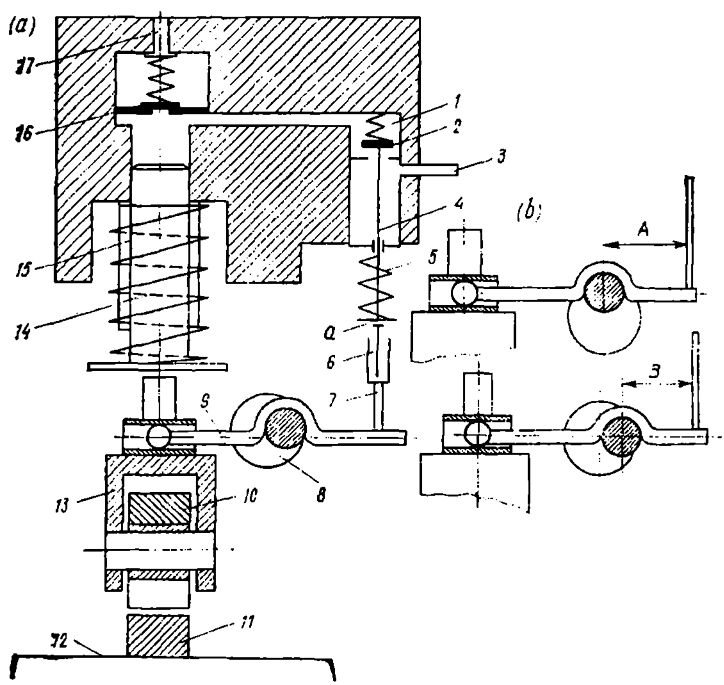


Fig. 63. Schematic of fuel-injection pump controlled by timed by-pass

is formed between upper tie-rod *4* and lower tie-rod *7* in spite of the action of spring *5* pressing the upper rod to the lower one.

In operation, as the lobe of cam *11* appears below roller *10*, lifting tappet *13* integrally with plunger *15* towards TDC, the fuel in space *1* is expelled into the suction line through open suction valve *2*. At the same time, the left end of cut-off lever *9* is also lifting while the opposite end is lowering, enabling the suction valve *2* to settle into its seat. From this moment and onward, plunger *15* is on the delivery stroke. As soon as the pressure of the fuel compressed by the piston exceeds the injector-opening pressure, the fuel is injected

into the engine cylinder over delivery valve *16* and high-pressure fuel line *17*. The delivery stops when plunger *15* reaches TDC. When the plunger starts its downstroke due to spring *14*, delivery valve *16* is closed by its spring and a vacuum is created above the plunger which enables suction valve *2* to be unseated and admit fuel from line *3*. At this stage, the clearance between the tie-rods is zero, permitting suction valve *2* to open.

It will be noted that in the pump controlled by timed by-pass, the quantity of fuel injected varies with the clearance a between the tie-rods. In fact, the earlier the suction valve *2* is in its seat, the more fuel is delivered into the engine cylinder. When $a = 0$ (the plunger is at TDC), all the fuel is by-passed into the line without reaching the injectors. The clearance is adjustable in two ways: by turning eccentric *8* of cut-off lever *9* or by changing the length of the tie-rods with the aid of screw *6*. Eccentric *8* is connected to the fuel regulation lever of the diesel engine. By operating this lever, eccentrics *8* of all the pumps of the engine are turned through the same angle. Consequently, the distance (*A* and *B* in Fig. 63*b*) between the eccentric centres and lower tie-rods *7* will change when the camshaft is set rotating. The greater this distance, the higher is the suction valve lift, and the later these valves will become closed. The period of delivering fuel to the injectors will be obviously extended in this case. To adjust the pumps for uniform delivery per cylinder, the recourse is to the tie-rods which must be adjusted for length; to reduce delivery, the length of the tie-rods is to be increased by turning screw *6* up and causing, consequently, suction valve *2* to close at a later moment of the cycle.

The delivery advance will change automatically with a change in the quantity delivered per cycle. If there is the necessity to change only the delivery advance, this is accomplished by changing the position of the cam on the camshaft. To advance the delivery, the cam is to be turned in the direction of camshaft rotation, for the cam lobe will appear under the tappet roller at an earlier time in this case. To retard delivery, the cam is to be turned in the opposite direction.

A point to be noted is that the fuel-injection pumps of the Д and ДР 30/50 diesel engines are actuated each by the same cam irrespectively of whether the engine is being run ahead or astern. This is achieved by fitting the cam to the camshaft so that the tappet roller is under the cam lobe at the instant when the crankpin is at TDC. Thus, the engine can be run both ahead and astern with the same delivery advance setting.

The pumps controlled by timed by-pass feature a simple plunger and deliver fuel in an amount varying with the engine speed. Therefore, they are employed on variable-speed engines, i.e. on those of the direct-coupled type. Since a decrease in engine load reduces the maximum compression pressure, these pumps ensure smooth engine

running. Their delivery advance changes automatically with the engine speed.

A schematic diagram of a fuel-injection pump with plunger-controlled ports is depicted in Fig. 64. Consider the operation of a single pump element shown at (*a*). The pump element consists of pump body *1*, pump barrel *8* with relief port *6* and inlet port *3*, plunger *5*, roller *10*, spring *9*, delivery valve *4* and control rack *2*. Plunger *5* performs a dual function, delivering fuel to the injector and effecting quantity control depending on engine speed. To that end, plunger *5* is provided at its top *1*, view *b*, with vertical passage *5* joining cut-off scroll *2*. When vertical passage *5* is set in front of the inlet port of the barrel, the fuel delivery is zero as this is shown in (*c*), view *3*. An intermediate position of the plunger, as shown in (*c*), view *2*, caters for a fuel delivery meeting the engine load, and a downstroke of the plunger causes fuel to enter the space above the plunger, (*c*), view *1*. Apart from being accurately machined, the plunger is lapped into the barrel. For a better sealing effect, the plunger may be provided with an annular groove *3* (Fig. 64*b*). Two arms *4* provided at the plunger bottom and engaging corresponding slots in a control sleeve which is a sliding fit on the barrel serve to turn the plunger about its vertical axis through an angle ensuring a given fuel delivery. Spring *9* (Fig. 64*a*) supported by a spring retainer provided at the top of the plunger returns it to BDC (Fig. 64*a*). Delivery valve *4*, functioning as a non-return valve, disconnects the high-pressure fuel line from the space above the plunger during the fuel cut-off. A cylindrical surface termed the shroud with a tail piece and longitudinal fuel passages, which are provided on the stem of the delivery valve, reduces the pressure in the injector line towards the end of delivery so as to prevent the dribbling of fuel.

In operation, when tappet *11* of the pump element is past the lobe of cam *12*, spring *9* causes the pump plunger to move towards BDC. The vacuum created above the plunger enables fuel to fill the space in the barrel through drillings *3* and *6*. When the lobe of cam *12* starts lifting roller *10* of tappet *11*, the plunger reverses the direction of its travel, moving towards TDC. Some of the fuel in the barrel is expelled into the suction line before the inlet ports are covered by the plunger. Further upward displacement of the plunger causes a pressure build-up to a point when the delivery valve is set open and fuel is delivered to the injector. With all the ports covered by the plunger, all the fuel contained in the barrel would be fed to the injector. However, at some point when the cut-off scroll uncovers the relief port, the pump chamber becomes connected to this port by way of the longitudinal passage. The fuel pressure sharply decreases, the delivery valve returns into its seat and the delivery comes to an end. The earlier the plunger uncovers the relief port, the less

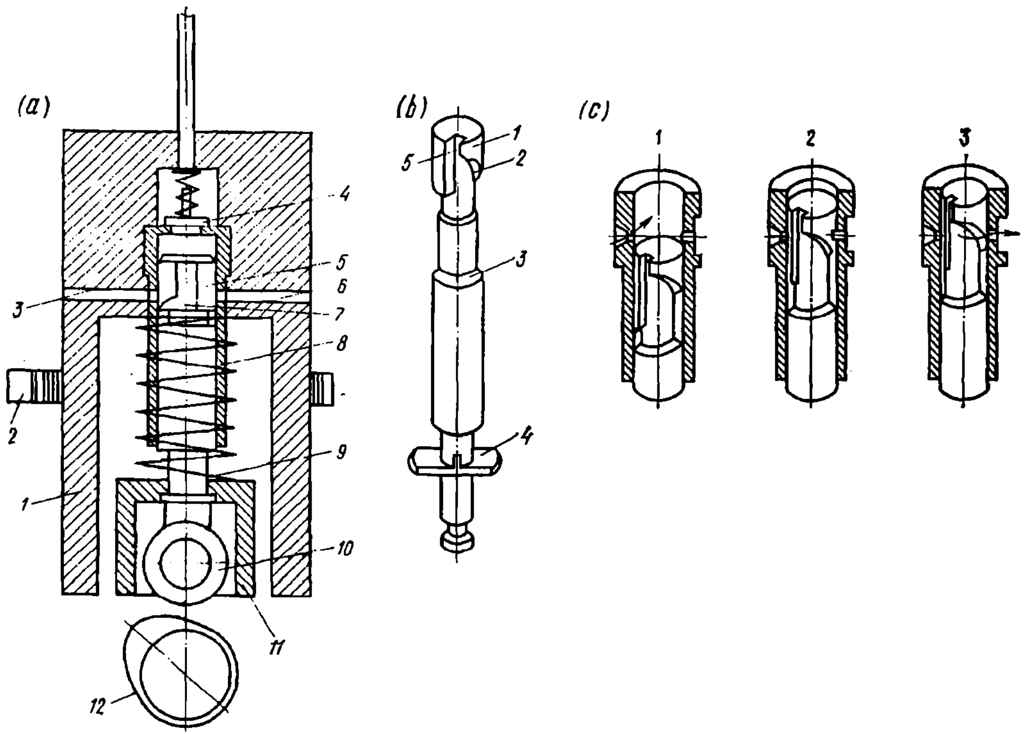


Fig. 64. Schematic of fuel-injection pump with plunger-controlled ports

fuel is delivered to the injector. Thus, the quantity of fuel injected per stroke is decided by the length of the plunger generatrix facing the relief port between the plunger top and cut-off scroll. In other words, the amount of fuel admitted into the engine cylinder can be regulated by changing the length of the plunger generatrix. This can be done by turning the plunger axially with respect to the barrel with the aid of the control rack *2* engaging a control quadrant attached to the control sleeve by a set screw. Apparently, meshing the control rack, which is linked to the engine control station, are the control quadrants of all pump elements. They are adjusted for uniform delivery by being turned with respect to the control sleeves.

The delivery advance is adjusted with a screw incorporated in the tappet, an earlier delivery being set by turning the screw up and a retarded one, by turning the screw down.

The pumps with the plunger-controller ports compare favourably with other types in that they snappily terminate delivery under a high pressure in the system and provide for good atomization of the fuel at rated engine load. Therefore, they are used mainly on diesel engines operating at a constant speed. Compactness, simplicity and trouble-free service are other advantages of these pumps.

Variable-speed diesel engines run inherently rough at low speeds, an increase in the combustion pressure at a high rate with a simultaneous extension of the ignition delay being the cause. To minimize the rough running of large direct-coupled diesel engines, steps are taken to shorten the ignition delay. This can be achieved by accelerating the plunger motion with the aid of cams having a sharply rising slope. Although this poses some manufacturing problems, including the selection of the right material for the cams, the advantages of the pumps with plunger-controlled ports offset the drawbacks, paving the way to their wide-spread application in marine practice.

56. Fuel Injectors

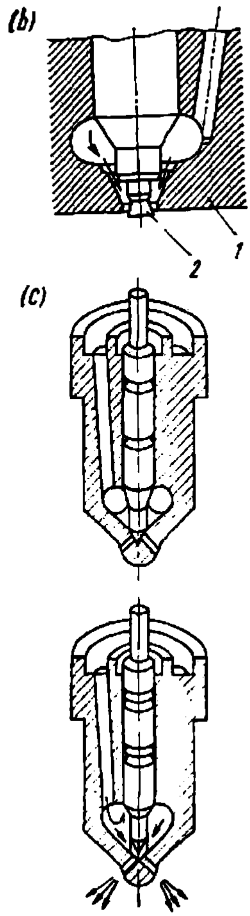
Injectors, also termed fuel-injection nozzles, serve the purpose of injecting fuel into the engine cylinders in a finely atomized spray. The fuel delivered by the fuel-injection pump under a high pressure is issued from the injector orifices measuring between 0.15 and 0.75 mm in diameter at a high velocity instrumental to the atomization. Fuel injectors exist in two varieties known as open and valve-closed.

Open injectors, i.e. those dispensing with a valve between the fuel line and combustion chamber, are not used on modern diesel engines. Although of a very simple design owing to the absence of moving parts, this type suffers from the afterdripping of fuel after the injection stroke. This fouls up the nozzle body and impairs combustion, inviting a loss of power.

A valve-closed injector (Fig. 65*a*) has hydraulically-operated

nozzle valve *3* separating delivery line *11* from the cylinder space and promoting efficient injection and combustion of fuel. Nozzle valve *3* fits into nozzle body *2* and is acted upon by spring *7* through spindle *5*. The spring tension is controlled with adjusting screws *8* and *9*. Once set for a given injection pressure, the adjusting screw is held fast by locknut *10*.

In operation, the fuel delivered by the injection pump enters an annular space in the nozzle body which is attached to holder body *6* by cap nut *4*. Acting on the valve head, the fuel pressure lifts the valve against the action of the spring through a distance of 0.4-1.2 mm. This clears the way for the fuel which is injected into the cylinder through orifices *1* of the noz-



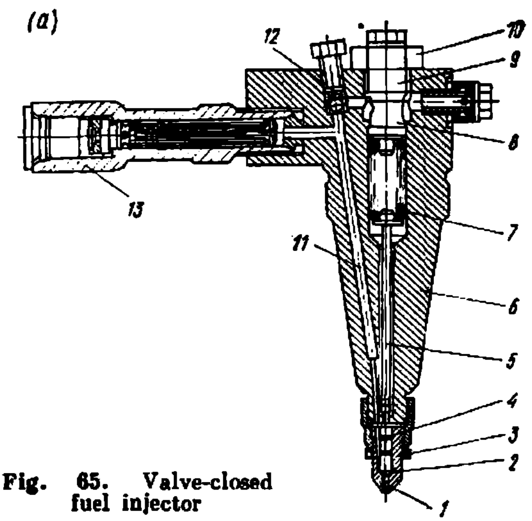


Fig. 65. Valve-closed fuel injector

zle body. As soon as the delivery ceases, relieving the fuel line of the pressure, spring *7* presses the valve against its seat so that the orifices become closed. The valve-opening pressure is a constant for the diesel engine of each particular type which is influenced neither by the engine load nor speed. It may be anywhere between $1\,370 \times 10^4$ and $3\,435 \times 10^4$ N/m^2 (140-350 kgf/cm^2) and is set by adjusting the tension of spring *7*.

Valve-closed injectors do not afterdrip and provide therefore for a non-smoky exhaust. Passage *12* is intended to empty the injector of fuel.

The fuel fed into an injector is cleaned there with the aid of edge-type filter *13* attached to the holder body. Referring to Fig. 66, the filter consists of rod *2*, milled in which are passages *1* and *3*, contained in case *4* with a clearance of 0.025-0.05 mm. The passages alternately connect to the spaces at the inlet into, and outlet from, the case, and the filtering is accomplished by pressure-feeding the fuel across the clearance so that the impurities with a particle size exceeding the clearance are trapped in the inflow passages.

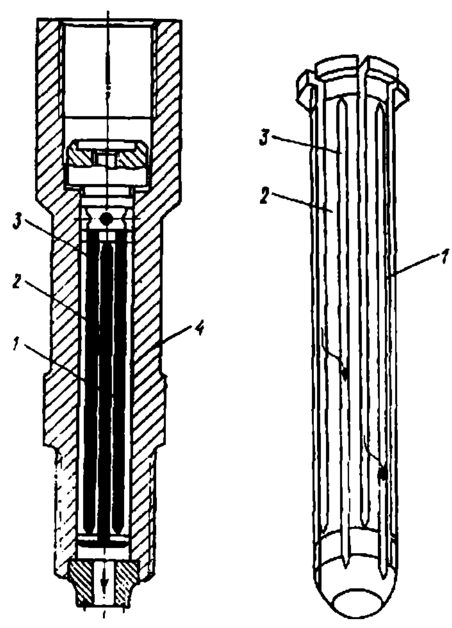


Fig. 66. High-pressure edge-type injector filter

For all the advantages of the valve-closed injectors which are apparent, this type is not free from some limitations. Before all, it is very sensitive to any impurity in the fuel. This may lead to valve seizure or broken spring which are defects rendering the injector inoperative.

Valve-closed injectors feature either single- or multi-orifice nozzle bodies. In the former (Fig. 65*b*), the nozzle valve has a cylindrical or taper pin at its end protruding from the orifice and forming an annular clearance with the nozzle body. In the latter (Fig. 65*c*), the plugging tip of the nozzle valve is given a taper shape. Single-orifice injectors find application on low-output diesel engines of the turbulence- or precombustion-chamber type. Multi-orifice injectors are used in direct-injection diesels. However, they may be clogged with sediments and carbon up in operation, displaying poor performance which results in incomplete combustion and excessive fuel consumption. Any fuel leaking past the nozzle valve into the nozzle body is drained via a leak-off pipe.

The nozzle valve and nozzle body are lapped to a very close fit, and in the case of wear or other defects the two parts cannot be interchanged individually. They are made of alloy steel.

To prevent a distortion of the holder body which may give rise to valve seizure, the nuts holding down an injector to the cylinder head are to be tightened in a uniform way by applying a controlled torque.

A trouble-free operation of injectors depends on the temperature of the nozzle valve and nozzle body which are bound to sustain the impact of hot gases. An overheating of these parts may lead to valve seizure and render the injector inoperative. Diesel engines with a bore over 400 mm commonly employ cooled injectors, the fuel digested by the engine being used as the coolant.

57. Injector Units

With an increase in diesel engine speed, the pressure waves set up in the fuel line connecting the injection pump to the injectors may be so strong as to interfere with the injection pattern prescribed by the profile of the pump-actuating cam. The elastic disturbance may cause extra lifts of the nozzle valve, poor atomization which lead to an excessive fuel consumption and produce a smoky exhaust. One of the ways of obviating the difficulties arising from pressure waves in high-pressure fuel lines is the elimination of these lines. This is achieved by combining the injection pump and injector in a single unit which is mounted on the cylinder head. Such a unit is termed injector unit.

Injector units have come into wide-spread application on two-stroke diesels with uniflow valve scavenging and on four-stroke high-speed units with a cylinder bore of 150-180 mm.

The fuel entering an injector unit from the fuel line is passed into an annular chamber of the plunger barrel from which it is injected through the orifices of the nozzle body, provided in the engine cylinder, under a pressure of $13\,700 \times 10^4$-$14\,700 \times 10^4$ N/m^2 (1 400-1 500 kgf/cm^2). Excess fuel is returned from the annular chamber into the fuel tank over an external pipe. An uninterrupted circulation of fuel through the annular chamber between the pump body and barrel promotes efficient cooling of the plunger with the barrel and facilitates the escape of air bubbles from the fuel system.

58. Principal Fuel Equipment Malfunctions

Leaky injectors and poor fuel atomization are defects which may develop during operation due to a number of causes, such as a broken injector spring, valve seizure, excessive wear of orifices or their clogging. Interfering with the intermixture of the air and fuel charges, they all provoke incomplete combustion and give rise to an excessive fuel consumption with a simultaneous loss of power. Although subjected to breaking are mostly springs with hidden flaws, a spring

which has been cocked in the process of fitting on the engine is also likely to break. A broken spring causes valve seizure and bad leakages of fuel. The seizing of the valve may also result from a solid particle between the nozzle valve and nozzle body. Fuel leakages are betrayed by a high exhaust temperature and valve seizures, by an overheating of the high-pressure fuel line.

Clogged injector orifices cause a sharp pressure buildup in the high-pressure fuel line, for the injection pump goes on delivering fuel to the injector. The pressure surge may break the pump plunger and rupture the high-pressure fuel line. Clogged injector orifices also impair atomization and the penetration of the fuel jet, this malfunction manifesting itself by a high exhaust temperature and smoky exhaust.

The seizing of the fuel-injection pump plunger leads to the misfiring of the cylinder and irregular crankshaft rotation. It can be pinpointed from the lack of throbbing in the high-pressure fuel line or from the temperature of the exhaust from each cylinder. If the plunger has become seized during a downstroke, the defect is indicated by the compressed spring.

An excessive wear on the pump plunger and barrel impairs the fit between the two parts, provoking heavy fuel leakages and reducing the amount of fuel delivered.

REVIEW QUESTIONS

1. What are the main components of the fuel system?
2. In which way is fuel separated from sediments?
3. What is the function of a fuel booster pump? What types of fuel booster pumps do you know? Discuss their operation.
4. What purpose does a fuel-injection pump serve?
5. Discuss the operation of an injection pump controlled by timed by-pass.
6. Discuss the operation of an injection pump with plunger-controlled ports.
7. What is the function of fuel injector?
8. Discuss the operation of a fuel injector.
9. What is the function of an injector unit?
10. What are the principal malfunctions of the fuel equipment?

Chapter X

THE LUBRICATING SYSTEM

59. The Function of the Lubricating System

Lubrication is vital for the operation of diesel engines in that it lessens friction and prevents excessive wear of the rubbing components, protects them against corrosion, removes some of the heat

resulting from friction, cools down the pistons, and prevents deposit accumulation.

Distinction is made between the following ways of feeding lubricant to the rubbing components of a diesel engine: full-force-feed lubrication of the bearings, splash lubrication, combination lubrication and metered lubrication by means of special pumps termed force-feed lubricators.

The pistons and cylinder liners of low- and medium-output engines are lubricated by oil mist formed by the splashing of oil dripping through the clearances in the main and crankpin bearings. Splash lubrication is also widely used in four-stroke medium-speed diesels with an output of 368-700 kW (500-950 hp) per cylinder. The rate of oil feed is decided by the rate of crankshaft rotation and the pressure of oil in the bearings in this case.

Simple as it is, the splash system of lubrication of the pistons and cylinder liners has a significant drawback: the lubricant reaching the rubbing surfaces is contaminated with the grit, which results from the wear on components, and the carbon entering the crankcase with gases.

The bearings of engines sustaining high loads are pressure-lubricated. In use are full-force-feed lubricating systems, also termed circulating systems, providing for a steady lubrication of the rubbing components along with the cooling of the pistons. The pressure of oil in the system for the lubrication of crankshaft and camshaft bearings, turbocharger bearings, auxiliaries drives, etc. is commonly of the order of 19.7×10^4-59×10^4 N/m^2 (2.0-6.0 kgf/cm^2) and up to 88×10^4 N/m^2 (9 kgf/cm^2) in some high-speed propulsion engines.

Metered lubrication is taken care of, as mentioned above, by multi-plunger pumps termed lubricators. Some of the trunk-type diesel engines use lubricators and splash lubrication (e.g. diesel engines of the Д and ДР 30/50 ranges). On crosshead engines, the pistons and cylinder liners are lubricated only by lubricators due to the presence of the diaphragm separating the crankcase from the cylinders.

For good results, the oil used in circulating systems must be of a grade suitable for given service conditions, for the bearing loads and temperatures in engines of various types are different.

A circulating system consists of an oil pump, lubricators, primary and fine oil filters, oil coolers. A relief valve (Fig. 67) is to be always incorporated into the system to by-pass some of the oil to the suction side of the pump or into the oil pan in the event of an excessive line pressure.

The bulk of the oil circulated through the system is contained either in the oil pan, also termed oil sump, or in a tank located in the engine room outside the diesel. In this latter case, the system is referred to as a dry-sump one. The "wet-sump" system is used on low-

and medium-speed engines and on some high-speed low-output ones. Apparently a very simple one, it is not free from limitations. For example, a limited capacity of the oil pan calls for recycling the oil at a rapid rate which shortens the oil change period.

The dry-sump system lessens the contamination and oxidation of oil, extends the oil change period. In this system there are two pumps, one drawing oil from the sump and returning it into the outside tank, and the other draws oil from the tank and delivers it to the engine on filtration and cooling. Apparently, the oil in the tank will deteriorate less than in the crankcase.

High-output marine diesel engines feature dry-sump lubrication without hazardous surges of the crankcase oil in the pan caused by the rolling or pitching of the ship in heavy sea. The pressure waves thus formed may damage the crankshaft or crankcase, let alone the fact that the suction line inlet laying bare may deprive the engine of lubricant with all the consequences.

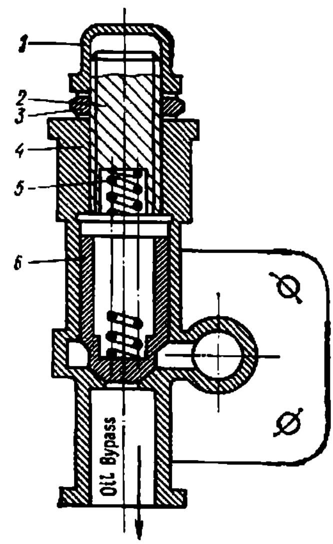


Fig. 67. Relief valve of lubricating system

1—cap; 2—adjusting screw; 3—lock nut; 4—valve body; 5—spring; 6—valve

Baffle plates provided in crankcases with "wet" sumps minimize the oil surges in stormy weather on some engines.

60. Typical Lubricating System Layouts

A "wet-sump" lubricating system is depicted in Fig. 68. The oil returned from all parts, and primarily the oil that was used to cool down the pistons, forms an oil sump at the bottom of the crankcase and drains therefrom into a drain tank located below the engine. A gear-type pump, drawing oil from the tank, supplies it to an oil cooler through a primary filter. The outflow from the cooler enters a main distributing passage which has branch lines feeding oil to the main bearings. Oil holes drilled in the crankshaft deliver oil to the crankpin bearings. The connecting rods are also provided with drilled oil holes extending through their shanks lengthwise over which oil is fed to the pistons to cool them down. Alternatively, telescopic pipes deliver oil to the pistons. Hence oil is returned to the crankcase where a wire net is provided above the sump to prevent frothing.

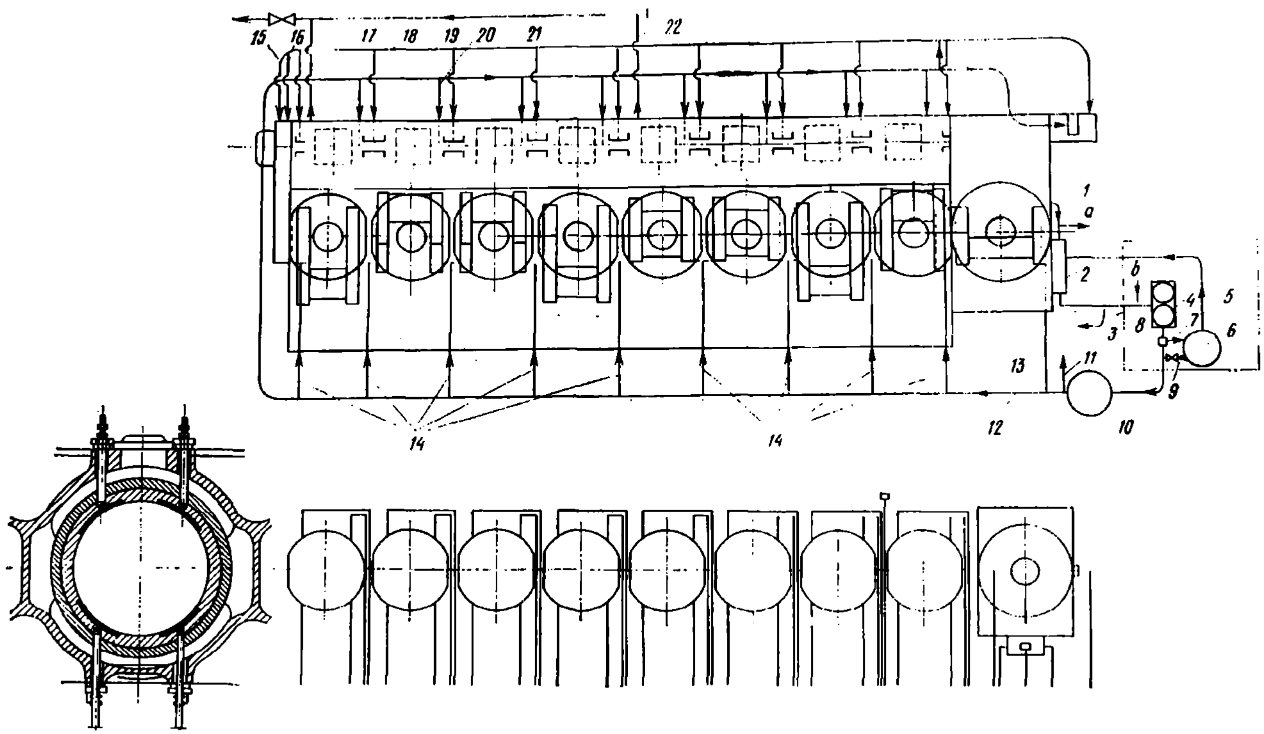


Fig. 68. Schematic of "wet" sump lubricating system

1—suction line from oil sump; *2*—oil pump; *3*—oil by-pass; *4*—oil strainer; *5*—outflow of filtered oil; *6*—fine oil filter; *7*, *9*—lines to and from fine oil filter; *8*—three-way valve; *10*—oil cooler; *11*, *15*, *16*, *17*, *20*—oil lines to auxiliaries; *12*, *13*—oil distributing passages; *14*—oil lines to main bearings; *18*, *19*—oil lines to camshaft bearings; *21*, *22*—return to oil sump

The camshaft and timing gear are lubricated by the oil fed from the main distributing passage over a branch line. A fine by-pass filter connected in parallel with the main distributing passage is incorporated into the system. The lubrication points of each cylinder served by lubricators can be seen in Fig. 68 (bottom, left and right).

For a more thorough cleaning of the oil from sediments, a centrifugal separator is connected into the main distributing passage as this is shown in Fig. 69 supplementing Fig. 68. The inflow into the separator is from the lower part of the tank via a preheater, and purified oil is directed into the oil sump. The capacity of the centrifugal separators used on medium- and high-output diesel engines is 500-5 000 l/h.

A stand-by electric pump is connected in parallel with the main oil-circulating pump for use during the starting and reversing of the engine. It is also used for draining used oil from the sump.

A dry-sump lubricating system is schematically represented in Fig. 70*a*. Its salient feature is the presence of two pumps, scavenging pump *11* and pressure pump *3*, operating in conjunction with outside tank *4*. The scavenging pump draws oil from the sump through strainer *12* and delivers into cooler *8* through primary filter *10*. The outflow from the cooler enters the outside tank from which pressure pump *3* feeds oil to the rubbing components over main distributing passage *1*. The system is fitted with relief valve *14*. The oil sump is provided with foam-depressing wire net *13*. Temperature controller *7* automatically maintains a constant oil temperature in the system by diverting some of the oil from the cooler into pipe *6*. The oil tank is equipped with an oil gauge and overflow pipe *5*. Partial flow fine filter *2*, circulated through which are 10-15% of the total amount of oil, is provided in the system. Preparatory to starting, oil is being run through the system with the aid of circulating pump *9*. A pressure gauge *M* and a thermometer *T* serve to monitor the performance of the system.

In an automatic lubricating system depicted in Fig. 70*b*, the return oil is collected in oil sump tank *13* before being taken care of by centrifugal separator *8*. Hence, the oil separated from sediments and water is delivered into supply tank *9* together with the oil gravity-fed from the oil sump. Fresh oil is admitted into the system through lines *IV* and *V*. Auxiliary electrical pump *10* connected into the line *V* feeds fresh oil into heater *12* the outflow from which, heated to at least 318 K (45°C), enters the system to ensure easy starting of the engine from cold.

For all the advantages offered by the dry-sump lubricating system, its layout is rather complex, featuring numerous lines, extra pumps and filters.

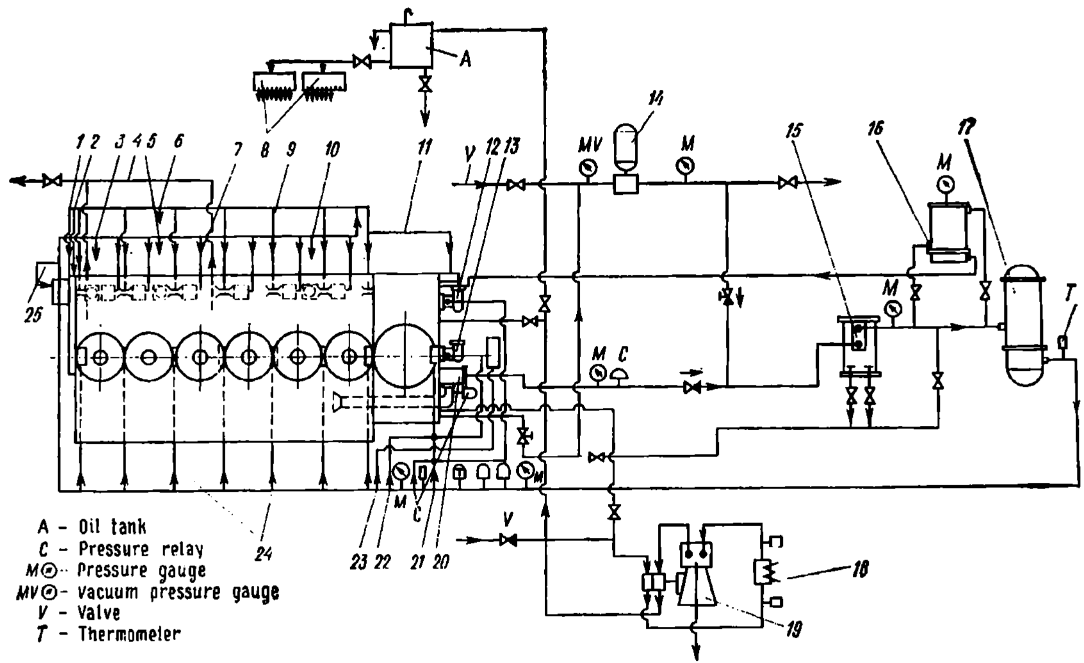


Fig. 69. Schematic of lubricating system featuring centrifugal separators

1—oil to camshaft timing gears; *2*—oil to idler gear journal; *3*—oil to fuel booster pump; *4*—oil return from receiver; *5*—oil to bearings of automatic starting gear; *6*—oil to automatic starting gear; *7*—oil to fuel-injection pump; *8*—lubricators; *9*—oil to camshaft bearings; *10*—oil to starting air-distribution valve; *11*—oil to camshaft end bearing; *12*, *13*—centrifugal water pumps; *14*—electric pump; *15*—strainer; *16*—fine filter; *17*—cooler; *18*—oil heater; *19*—centrifugal separator; *20*—oil pump; *21*—oil to outside main bearing; *22*—oil to centrifugal water pumps; *23*—oil to vibration absorber; *24*—oil to main bearings; *25*—oil to governor drive gear

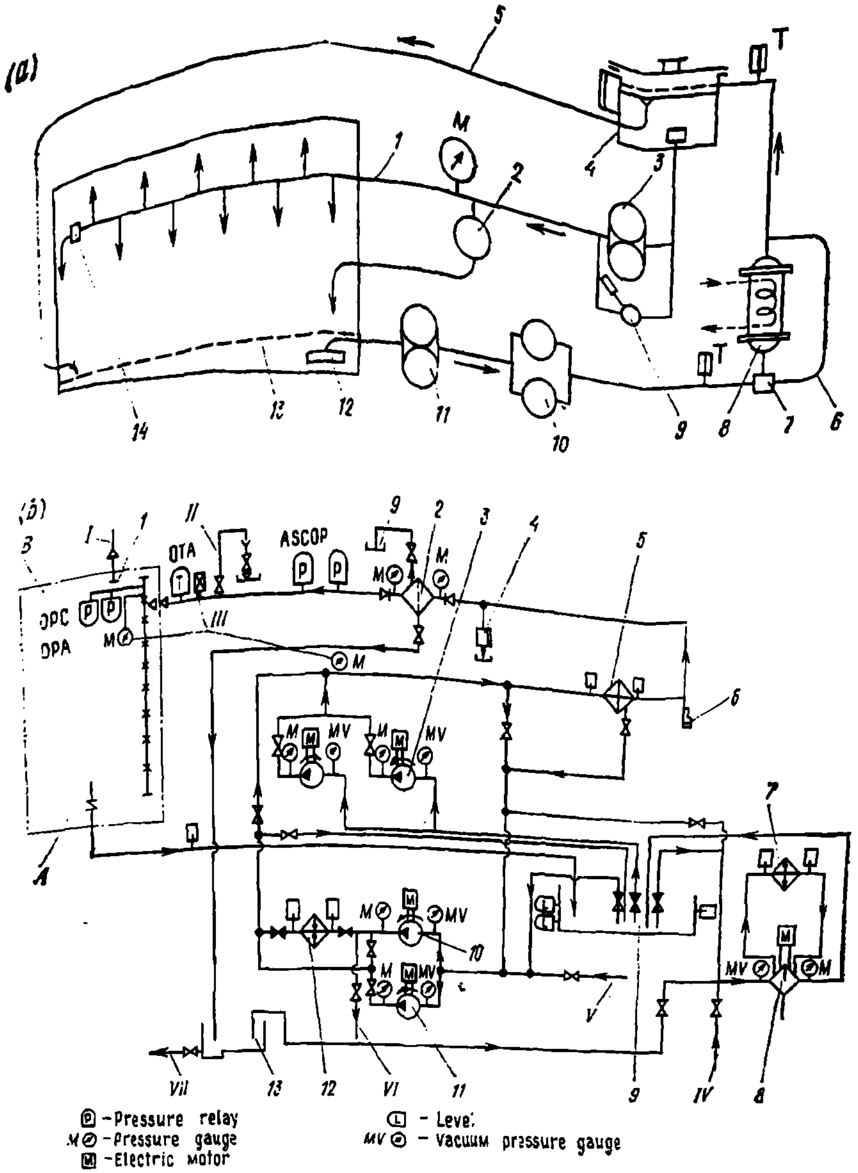


Fig. 70. Schematic diagrams of lubricating systems

(a) dry-sump type:

1, 6—oil lines; *2*—fine oil filter; *3*—pressure oil pump; *4*—oil tank; *5*—overflow pipe; *7*—temperature controller; *8*—oil cooler; *9*—pump; *10*—strainer; *11*—scavenging oil pump; *12*—strainer at inlet; *13*—foam-depressing wire net; *14*—reduction valve

b) automatic type:

1—diesel engine; *2*—oil filter; *3*—main electric oil pump; *4*—relief valve; *5*—oil cooler; *6*—temperature transducer; *7*—oil preheater before centrifugal separator; *8*—centrifugal separator; *9*—supply tank; *10*—auxiliary electric oil pump; *11*—standby electric oil pump; *12*—oil heater; *13*—oil collecting tank

I—to atmosphere; *II*—pipe for taking test samples of oil; *III*—to remote control panel; *IV*—clean oil inlet; *V*—standby clean oil inlet; *VI*, *VII*—outflow of oil; *OPC*—low oil pressure protection ($p \leqslant 35 \times 10^4$ N/m² or 3.6 kgf/cm²); *OPA*—low oil pressure alarm ($p \leqslant 45 \times 10^4$ N/m² or 4.6 kgf/cm²); *ASOP*—automatic starting of circulating oil pump ($p < 48 \times 10^4$ N/m² or 4.9 kgf/cm²); *OTA*—high oil temperature alarm ($t \geqslant 63$°C); *ASCOP*—automatic shutting down of circulating oil pump ($p > 55 \times 10^4$ N/m² or 5.6 kgf/cm²); *A*—fore end of diesel engine; *B*—aft end of diesel engine

61. Oil Pumps

Oil pumps used in diesel engines to circulate oil are of the gear and screw type. Consider the design features of a reversible single-section oil pump with automatic valves (Fig. 71).

Power impeller gear *4* and idler impeller gear *7* are accommodated in pump body *1* with suction and delivery valves, the power impeller gear being driven by a crankshaft gear. When the impeller gears are set rotating, a suction area and a delivery area are formed at the

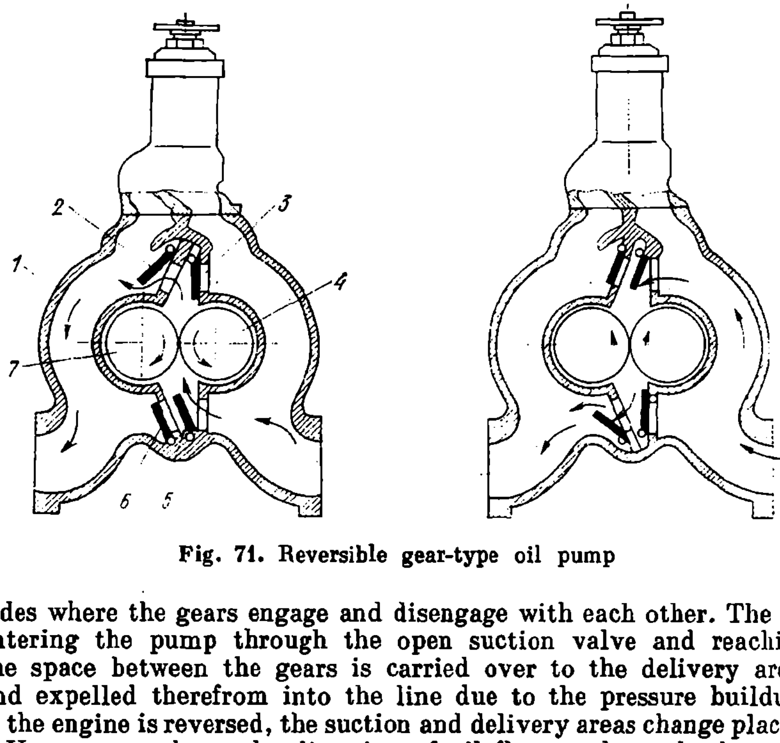


Fig. 71. Reversible gear-type oil pump

sides where the gears engage and disengage with each other. The oil entering the pump through the open suction valve and reaching the space between the gears is carried over to the delivery area, and expelled therefrom into the line due to the pressure buildup. If the engine is reversed, the suction and delivery areas change places.

However, to keep the direction of oil flow unchanged when the engine is being reversed, the pump is provided with automatic valves and a by-pass canal. When the power impeller gear rotates clockwise, the suction side is located next to two upper valves *2* and *3* with the result that valve *3* located to the right opens, admitting oil to the impeller gears which reaches them through the by-pass canal; left-hand valve *2* is pressed against its seat at this stage. The pressure at the delivery side opens lower left-hand valve *6* while right-hand valve *5* remains closed. When the impeller gears change the direction of their rotation, the suction and delivery sides change

places with the result that oil enters the pump through lower right-hand valve *5* and is delivered through open upper left-hand valve *2* while valves *6* and *3* are closed. It will be noted that the direction of oil flow has not changed in this case. In a non-reversible pump (Fig. 72), the valves are replaced by blanking-off plates *1* and *2*.

To keep the oil pressure in the system at a given level, the pump is provided with relief valve *3* adjusted to open against the action of its spring under a certain pressure which can be set with the aid

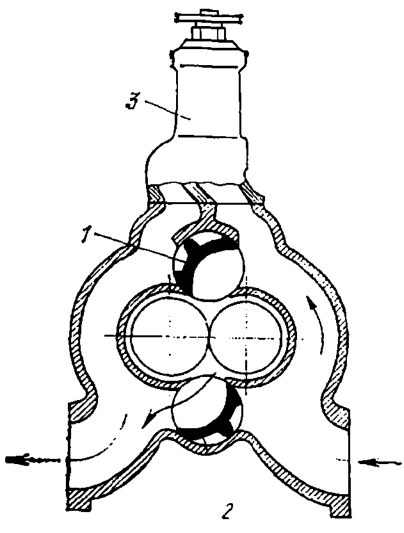


Fig. 72. Non-reversible gear-type oil pump

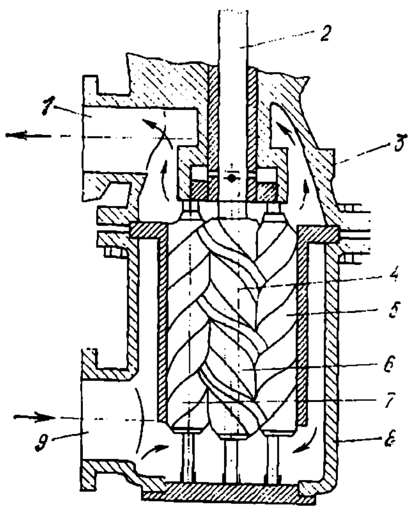


Fig. 73. Screw-type pump

of a screw changing the spring tension. If the line pressure exceeds the valve-opening pressure, the relief valve opens and by-passes excess oil into the suction side. The relief valve operates, for example, when the engine is being started from cold and the lubricating system is filled with cold oil displaying high viscosity. The same occurs when some of the oil filters become clogged.

Gear pumps have low capacity and are therefore widely used in low- and medium-output diesel engines in spite of noisy operation and pressure pulsation. They are also very simple and have a volumetric efficiency of 0.85-0.9. Gear pumps are compact units suitable for mounting on the engine directly. To boost capacity, use is made of duplex and triplex oil pumps, i.e. pumps consisting of two or three sections each fitted with a pair of impeller gears.

High-output diesel engines with dry sump employ screw oil pumps

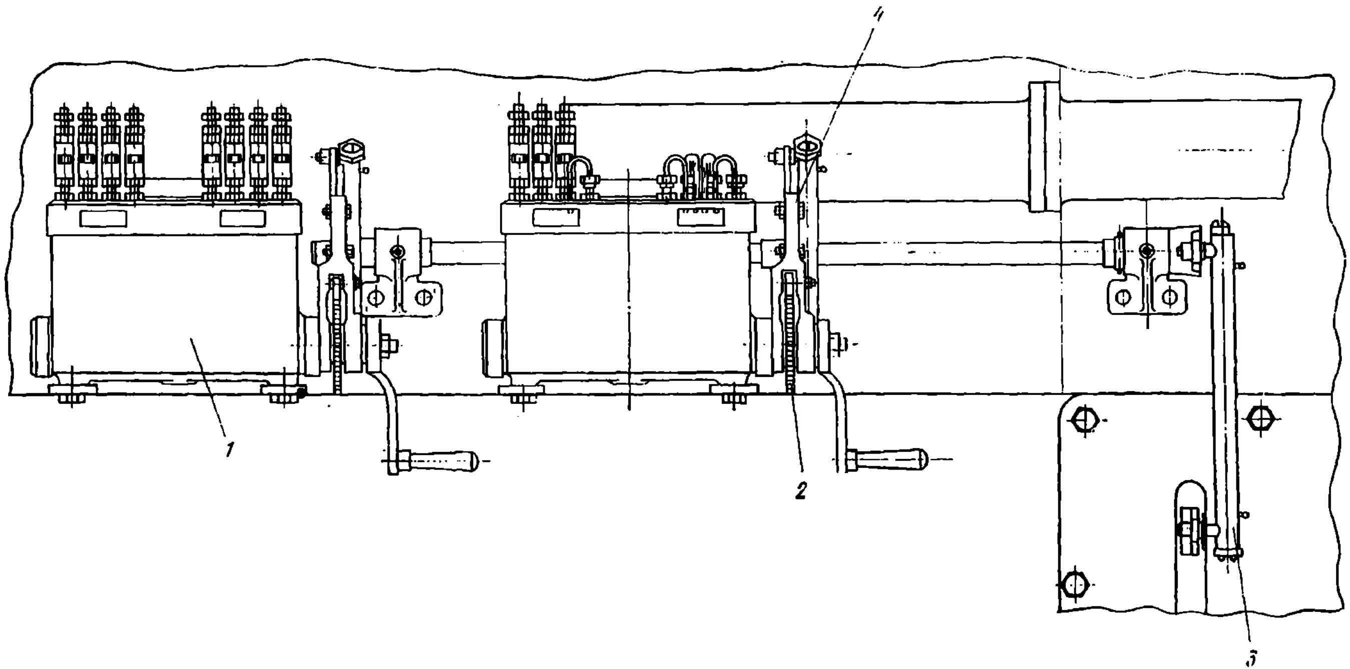


Fig. 74. Lubricators

with a separate electric drive. These high-capacity units capable of producing a significant suction lift are trouble-free in service. They are also noiseless and ensure the delivery of oil over long lines, being thus particularly suitable for large low-speed diesel engines with a high power output.

Referring to Fig. 73, a screw pump incorporates central helix *6* meshing with one or two idler helices *5* and *7* on parallel shafts which are contained in casing *8*. The rotation of drive shaft *2* causes the oil in suction side *9* to enter the spiral grooves of the intermeshing helices and flow towards delivery side *1*. Screw pumps are all of the non-reversible type. Apart from the difficulties experienced in generating the helical surfaces of the impellers, their only limitation is significant pumping loads tending to displace the helices towards the suction side. The thrust can be eliminated by using left- and right-hand helices which balance the pumping load. Some of the thrust is also eliminated by using dummy piston *3* in conjunction with axial passage *4*.

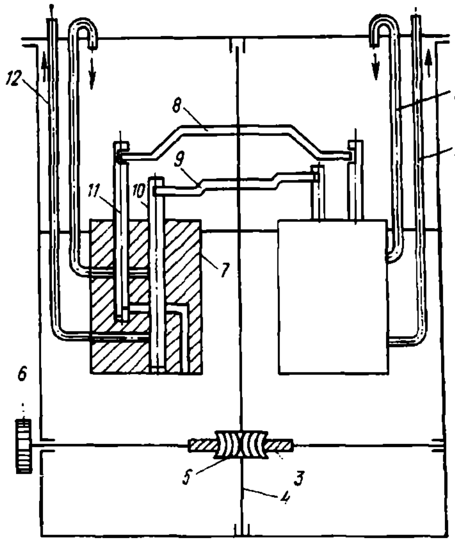


Fig. 75. Schematic diagram of lubricator

For a metered delivery of lubricating oil to the liners of engine cylinders at regular intervals, the recourse is to multiplunger oil pumps termed lubricators *1* (Fig. 74) driven from the camshaft or scavenging pump *3* by means of lever *4* and pawl-and-ratchet *2* or a reducer. A lubricator (Fig. 75) consists of an oil-filled casing containing means of oil delivery in the form of concentrically-arranged pump elements *7* varying in number depending on the lubricator type and actuated each by drive *3*, *5*, and *6*. Specially shaped discs *8* and *9* attached to vertical shaft *4* convert the rotary motion of the drive into a reciprocating motion of plungers *11* and control valves *10* of the pump elements, respectively. One revolution of disc *8* causes two double strokes of plungers *11*, one stroke being a delivery stroke feeding oil through lines *2*, *12* and the other stroke being a monitoring stroke feeding oil through pipes *1* serving to monitor the oil supply. The outflow from pipes *1* returns into the oil bath. Each revolution

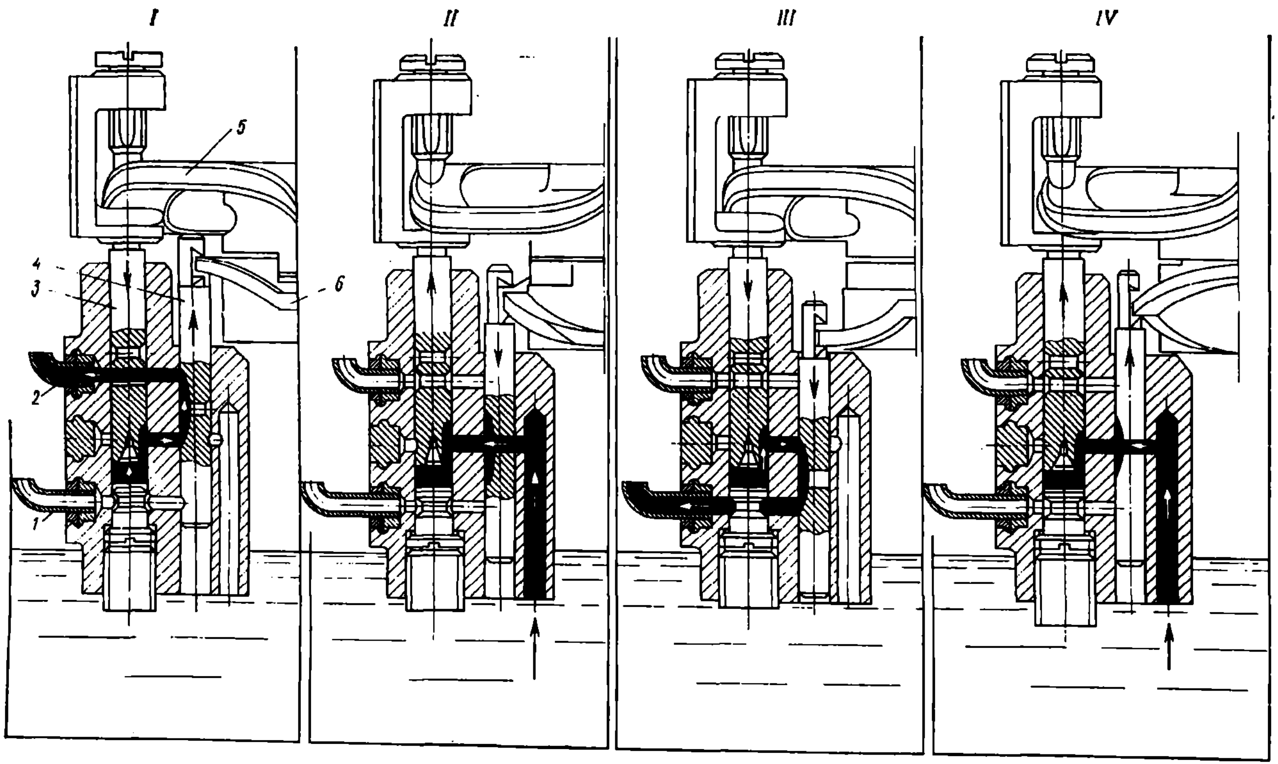


Fig. 76. Operating sequence of lubricator

1—delivery pipe; *2*—monitoring pipe; *3*—plunger; *4*—control valve; *5*, *6*—plunger-actuating levers; *I*, *III*—delivery strokes; *II*, *IV*—suction strokes

of disc *9* causes control valves *10* to perform an upstroke and a downstroke. For details of lubricator operation see Fig. 76.

Before starting a diesel engine, the oil is to be circulated by rotating the ratchets, using hand grips. The amount of oil fed per stroke can be adjusted by setting the effective plunger stroke with the aid of a tappet-adjusting screw: the screw rotating clockwise increases oil delivery and vice versa.

62. Primary and Secondary Oil Filters

For a thorough purification of oil from the products of its decomposition and sediments, primary and secondary filters are incorporated into the lubricating system.

Primary or coarse filters referred to in marine practice as strainers are intended to remove the coarser impurities from the oil. They all are high-capacity contrivances of the internal-surface type identical to fuel filters in point of design. Depicted in Fig. 77 is a primary wire gauze filter consisting of two self-contained sections *1* and *2* which are connected to the system alternately so that the idle section can be cleaned without stopping the engine. Each section consists of a filter pack which is a stack of wire gauze filtering elements with the meshes measuring 0.14 × 0.14 mm and upwards. The stack is clamped by discs and contained in a case. On being admitted into the case via a three-way valve, oil passes through

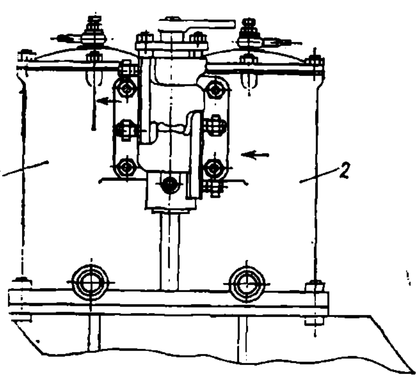


Fig. 77. Primary wire gauze filter

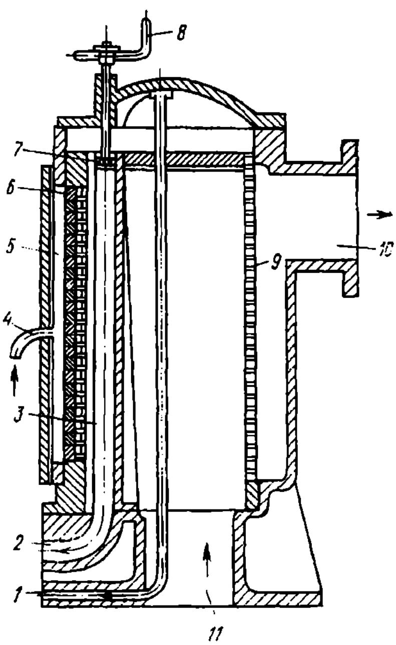


Fig. 78. Burmeister & Wain self-cleaning oil filter

the filtering elements and enters a central tube connected to a chamber at the case top from which the oil is taken to the outflow line. The sludge consisting of sediments and water is accumulated in the lower part of the case and is drained through a cock; an air bleeding cock is provided in the cover of the case. Two pressure gauges, one connected downstream and the other upstream of the strainer, or a differential pressure gauge, are used to monitor the strainer performance. A pressure difference of 14.7×10^4 N/m^2 (1.5 kgf/cm^2) indicates that the strainer is clogged and needs cleaning.

To save time wasted in cleaning and washing filters, use is made of semi-automatic and automatic purification facilities. A self-cleaning Burmeister & Wain filter (Fig. 78) incorporates wire gauze filtering drum *9* with 2×2 mm and 0.25×0.25 mm meshes rotatable by hand with the aid of hand grip *8*, pinion *7* and a gear ring attached to the drum. A stack of filtering elements *6* is also provided in the filter case next the filtering drum. The inflow of oil is via pipe *11* and purified oil leaves the filter by way of pipe *10*. For cleaning the filter, compressed air is fed through union *4* into space *5* and hence inside filtering drum *9* and through the wire gauze. The impurities collecting in space *3* are disposed of by the air flow into a collecting tank over passage *2*. Simultaneously with admitting the air into the filter, the filtering drum is being rotated by hand grip *8*. Air is vented from the filter through pipe *1*.

Strainers are supplemented by magnetic filters which remove the grit from the oil which results from wear on components.

In use are also edge-type primary filters comprising a stack of perforated discs separated by star-shaped spacers which form spaces between 0.01 and 0.07 mm wide for filtering the oil. Purified oil flows into the line through the disc perforations. By rotating a handle connected to the shaft stacked on which are the filtering discs, thin stationary fingers extending between the discs scrap off all dirt between them which falls to the bottom of the case and is disposed of through a cock. Edge-type filters are reputed for good performance but display a low capacity and are used therefore on low-output diesel engines.

Secondary or fine filters are incorporated into the lubricating system as adjuncts of the primary ones. They all fall under the category of what is called cavity filters and are capable of separating particles between 0.01 and 0.001 mm. However, their capacity is commonly low and the resistance rather high. The usual practice is to pass through the fine filters about 7 to 15% of the oil circulated through the system. To that end, they are connected in parallel with the strainers and are referred to as by-pass or partial-flow filters.

Partial-flow fine filters have been used as standard equipment on low- and medium-output diesel engines. Available nowadays are

full-flow fine filters of the Narva type which are simple and low-cost units capable of affording fine-particle filtration. The filtering elements are made of nonwoven bonded fabric or long-fibred paper and give a service life of 2 000 h. The test carried out on some diesel engines have proved that the full-flow fine filters of the Narva type reduce the wear on cylinder liners by 32%, on piston rings by 28%, on bearings by 48%; they also reduce the carboning up of components and extend the service life of engine components by at least 20%.

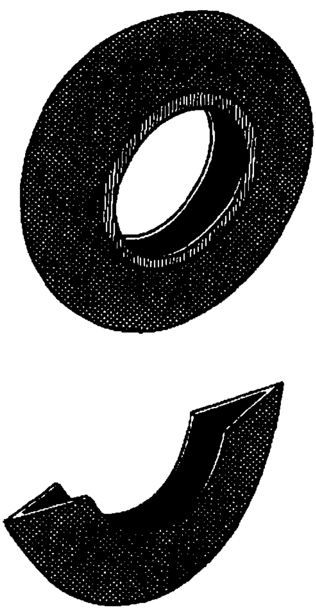


Fig. 79. Replaceable filtering element in paper

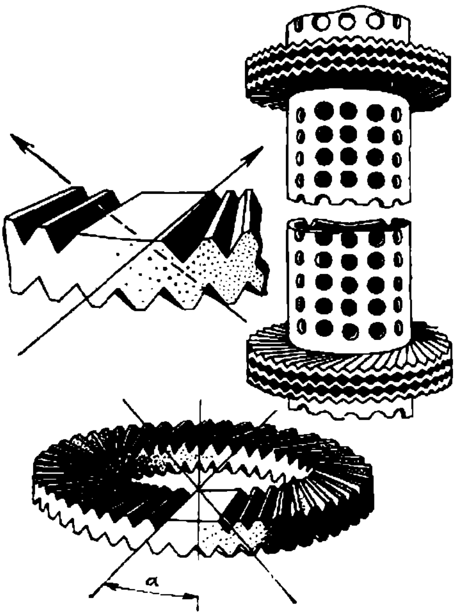


Fig. 80. Fipoca filtering element

Full-flow fine filters are now standard equipment of the 6ЧН 21/21, ЧН 26/26, 6Н 15/18, 16Ч 26/26 and some other diesel engine batch-produced in this country.

Many ships employ the purification of lubricating oil in centrifugal separators used in conjunction with filters.

Among other fine filters designed to meet stringent dirt-holding requirements to ensure continuous and trouble-free operation of high-powered medium- and high-speed diesel engines there are units with replaceable paper elements (Fig. 79) of various kind which are self-cleaned by pressure-fed oil. Pielstick diesel engines employ Rellumix filters with Fipoca elements. Each element is a disc made of superpolyamide or metal and having corrugated surfaces. The

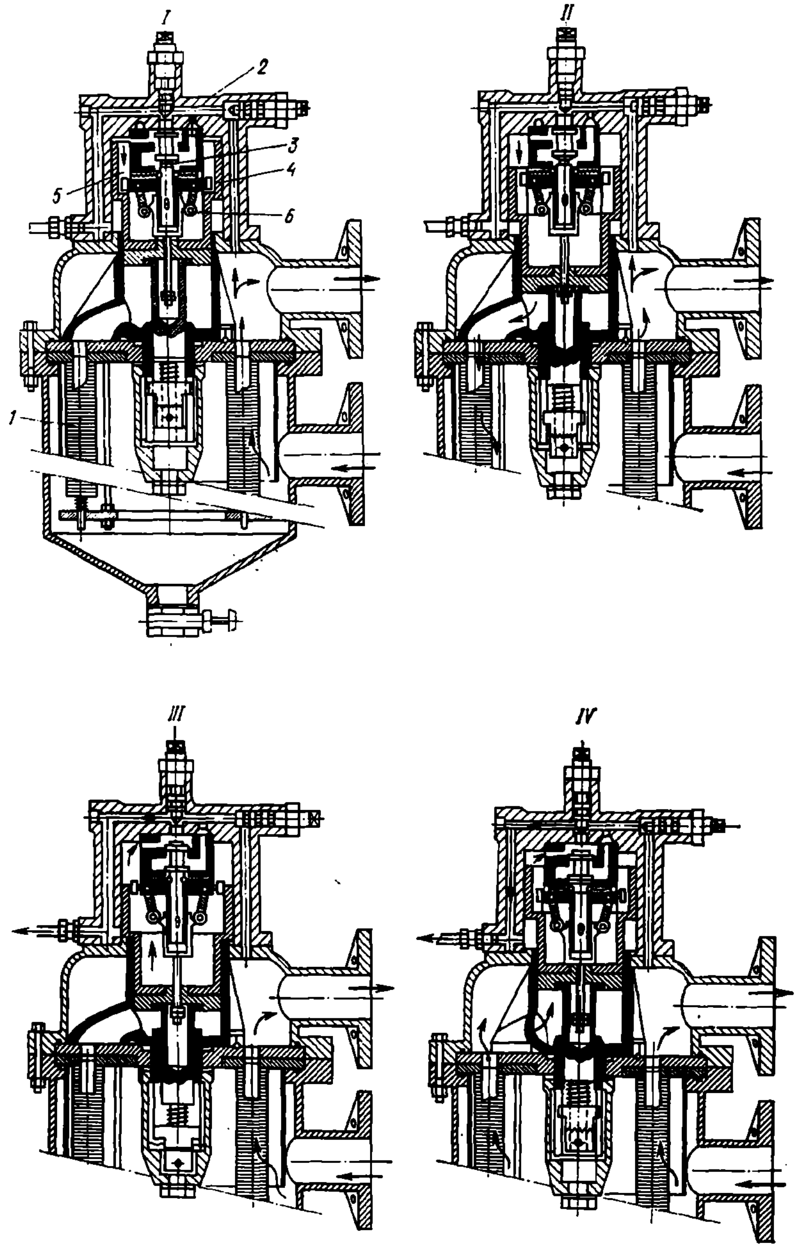


Fig. 81. Rellumix automatic oil filter

radial troughs between the ridges form oil ways of a triangular cross section increasing in size in the direction of oil flow (Fig. 80). A pack of filtering Fipoca elements assembled on a perforated cylinder forms a cartridge used to filter oil. The filter capacity varies with the number of filtering elements. The oil ways have a grade preventing

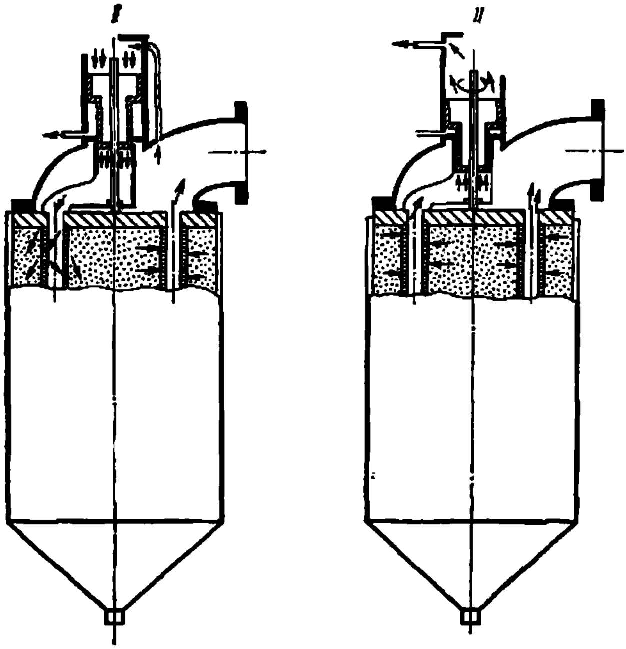


Fig. 82. Operating sequence of Rellumix oil filter

an ingress of the impurities into the central space of each cartridge along with the oil which is pumped therefrom into the main distributing passage.

A Rellumix automatic filter (Fig. 81) consists of lower case *1* containing Fipoca filtering elements and upper case *2* accommodated in which are differential piston *4*, spool valve *3*, rotary feeder *5* and pawl-and-ratchet mechanism *6*. The self-cleaning action consists in that some of the filtered oil enters the feeder chamber and a space above the differential piston, causing a differential pressure buildup which displaces the piston downward (Fig. 82). The resulting reversal of the oil flow cleans the filtering elements from the impurities settled on their external surfaces when the packs are alternately set below the feeder by the pawl-and-ratchet mechanism attached to the piston rod.

This filter affords continuous full-flow filtration without much attendance at a rate of up to 300 m^3/h in spite of being a compact unit. However, its construction is complex, and the filtering packs need to be hand-washed in a special solvent at regular intervals.

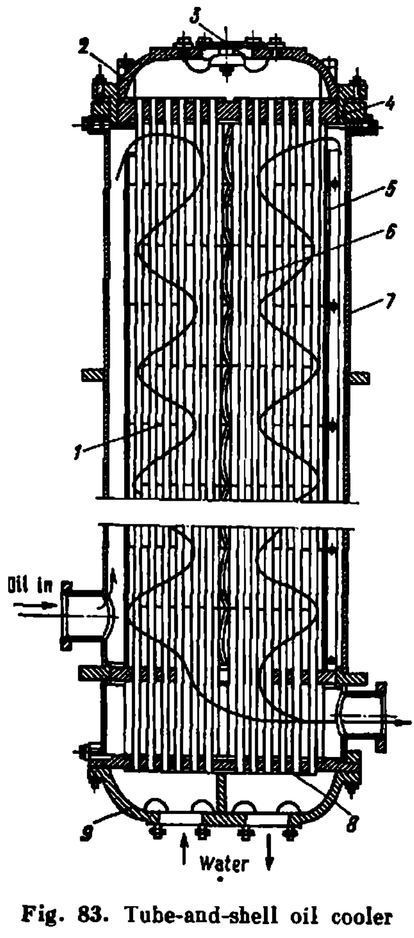


Fig. 83. Tube-and-shell oil cooler

63. Oil Coolers

Bound to operate at the temperatures of 333-353 K (60-80°C), lubricating oils will give unsatisfactory performance due to their low viscosity and poor oiliness unless they are cooled down before being recirculated through the engine. To that end, oil coolers are incorporated into the lubricating system. They are classified with reference to the design features into shell-and-tube and plate coolers.

Most of the oil coolers in shipboard use belong to the shell-and-tube type, because they meet the requirements intended to ensure trouble-free operation and efficient cooling in the most satisfactory way. A tube-and-shell cooler illustrated in Fig. 83 comprises casing *7*, shell *5*, tube plates *2* and *8*, copper tubes *6*, upper shell cover *3* and lower shell cover *9*. The open ends of tubes *6* are expanded into the tube plates. Upper tube plate *2* is of the floating type, being thus capable of displacing due to the thermal elongation of the tube bundle in operation. To prevent oil leakage along the casing, stuffing box *4* consisting of a packing held fast by a gland is fitted at the upper tube plate. An annular clearance is provided between casing *7* and shell *5* surrounding the tube bundle. Cooling water is circulated through the tubes. Oil is admitted into the annular space at the bottom of the cooler, rising then to the top where it reverses the direction of travel and descends to the bottom in a counter-current

flow with respect to the water. Transverse baffle plates *1* extend the oil path through the cooler. The upper shell cover is provided with a zinc corrosion inhibitor safeguarding cooler components against the corrosive effect of the cooling water. For trouble-free operation, the tubes must be cleaned at regular intervals.

Plate oil coolers also find application on some ships. An Alfa Laval plate cooler (Fig. 84) consists of a frame in which independent

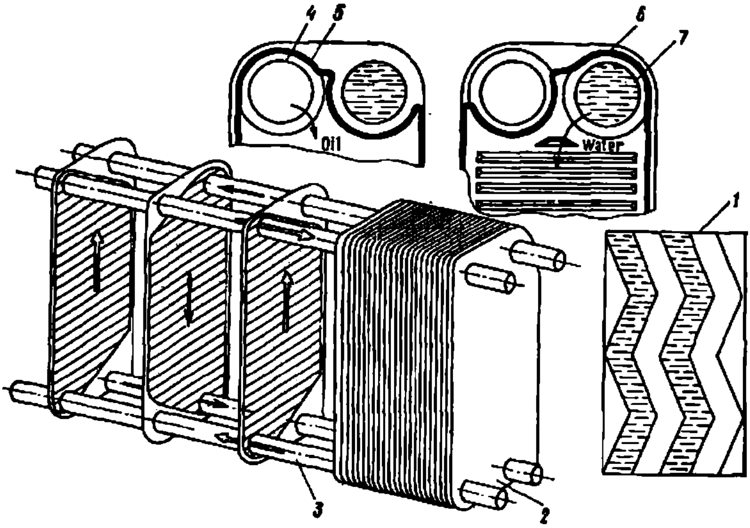


Fig. 84. Alfa Laval plate oil cooler

metal plates *2*, each 1.2 to 1.4 mm thick, supported by tubular rails *3* are clamped between a head and a follower by means of bolts. The plates are referred to as first and second ones and are provided each with four ports at the corners, the former having their oil or water ports located in the left-hand and the latter in the right-hand corners. The plates are sealed at their outer edges and around the ports by gaskets *5* and *6* made from a heat-resistant rubber capable of standing a temperature of 393-413 K (120-140°C). The gaskets are so arranged that the oil and the cooling water are directed alternately into the passages formed between the plates, flowing there counter-currently. The plates have horizontal troughs which add to their stiffness and produce turbulence in the liquids flowing between the plates in slow thin streams of large area owing to inclined corrugations *1*.

Plate oil exchangers are compact units readily accessible for maintenance. However, they need more attention in operation than the shell-and-tube type, particularly as far as the pressures of the

oil and water flows are concerned which must be kept equal to prevent the deformation and cracking of plates. Plate oil coolers may be sometimes too long and need adequate space for their installation.

64. Centrifugal Separators

The problem of separating water, organic matter and sediments from lubricating oil (or fuel) is coped with in marine practice by installing centrifugal separators of various construction on board ships.

Consider the construction of a vertical continuous disc separator (Fig. 85). The separator consists of frame *7*, step-up gear *4*, bowl *6* and dry collecting housing *5*. The bowl is the key component in which all the impurities contained in oil are separated from it. The step-up gear of the worm type obtains the drive from electric motor *1* via friction clutch *2* connected to horizontal shaft *3* and causes the bowl to spin at 5 000-8 000 rpm integrally with the shaft it is fitted to.

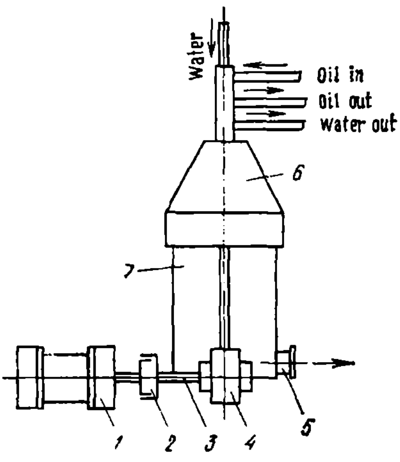


Fig. 85. Continuous disc-type centrifugal separator

For operation, the separator relies on the centrifugal force set up in the bowl which causes the sediments and water to migrate towards the periphery of the bowl, for they are heavier than oil, while the oil migrates towards the centre. The water overflows into an annular space between the topmost disc and the conical separator cover, being then disposed of. The sediments, on passing across a set of conical discs, are thrown off into the dry collecting housing. The purified oil rises to the top, leaving then into a tank through an annular space between the water outflow pipe and the disc retainer.

Distinction is made between two modes of separator operation which are termed purification when both water and sediments are removed from oil (Fig. 86*a*) and clarification when taken care of are sediments alone (Fig. 86*b*). Residual fuels are commonly treated for purification which is a versatile method of fuel conditioning.

To speed up the process of centrifugation, fuel is being heated up by introducing hot water (its temperature is 3 to 5°C higher than that of fuel) into the separator in an amount of 3 or 4% of the fuel treated. The hot water also removes soluble salts and ash from

the fuel. The dry collecting housing is to be cleaned from sediments at regular intervals to ensure trouble-free functioning of the separator.

In use nowadays are self-cleaning separators discharging the impurities at regular intervals without being taken apart. They

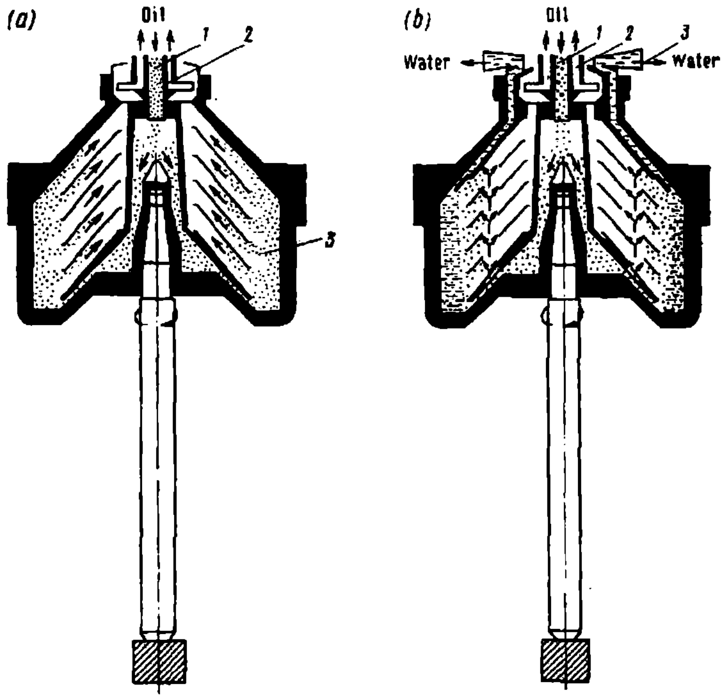


Fig. 86. Operating modes of separator
(a) purification:
1—contaminated oil in; 2—purified oil out; 3—sediments
(b) clarification:
1—contaminated oil in; 2—clarified oil out; 3—impurities out

are programme-controlled and capable of initiating visual and audible alarms before shutting off automatically the inflow of oil or fuel in the event of a malfunction.

To facilitate the centrifugation, the fuel is preheated to a temperature of max 363 K (90°C) in an electric heater. The self-cleaning procedure is explained in Fig. 87. Preparatory to cleaning, sliding bottom *1* of the bowl is pressed against the stack of discs due to the pressure of water as shown in (*a*). In the course of cleaning the bottom is relieved of the water pressure, sinking therefore downwards.

The sequence of events during the discharging of the sediments is as follows:

—the flow of oil into the separator is discontinued and the remnants of oil are expelled from the bowl by the flushing water;

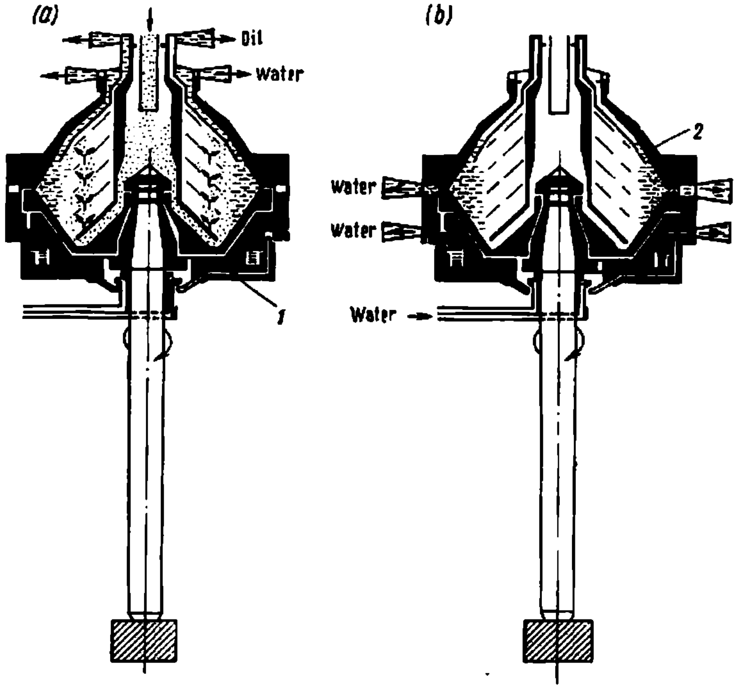


Fig. 87. Operating sequence of self-cleaning separator
(1) bottom of bowl pressed against disc stack due to water pressure; (2) water drained from under bowl bottom

—water is admitted into the hydraulic system to open the spring-loaded valves provided in the bottom of the bowl (*b*);

—the draining of water from below the sliding bottom depriving it of support causes an abrupt separation of the bottom from the stack of discs so that the impurities are thrown out of the separator due to the centrifugal force overcoming the adhesive force.

The cleaning is carried out without stopping the separator.

65. Malfunctions of the Lubricating System

For trouble-free operation of a propulsion plant, the main thing to look out for the personnel is to maintain normal oil pressure and temperature in starting and running the engine.

Factors that cause low oil pressure or discontinue oil supply are as follows:

—low viscosity of the oil due to its dilution with fuel or water or an increase in the oil temperature in excess of an allowable maximum; the remedy is to change the oil immediately if a test proves that it is unsuitable for further use;

—high resistance to the flow of oil through coolers or filters; washing the filters or cleaning the oil coolers will eliminate the defect;

—leaky oil delivery line or weak spring of the relief valve; leaks must be detected and eliminated or the tension of the relief valve spring must be increased;

—defective valves of the reversible oil pump; the pump must be taken apart and the defective valve rectified;

—too wide clearances in main, crankpin and piston pin bearings; setting correct clearances will restore normal lubricating conditions.

REVIEW QUESTIONS

1. What purpose does lubrication serve?
2. What lubricating systems are in use on diesel engines?
3. Describe the operation of the "dry-sump" and "wet-sump" systems of lubrication.
4. What types of oil pumps are used on diesel engines?
5. Discuss the operation of a reversible gear-type oil pump when the diesel engine is running "AHEAD" and "ASTERN".
6. What is the purpose of fitting diesel engines with high-pressure multi-plunger oil pumps known as lubricators?
7. How is a pump element of a lubricator functioning?
8. How are sediments separated from lubricating oil?
9. What purpose do oil coolers serve?
10. What types of oil coolers are used on diesel engines?
11. What malfunctions of the lubricating system are met with in running a diesel engine?

Chapter XI

THE COOLING SYSTEM

66. The Function of the Cooling System

Out of the great amount of heat liberated in the engine cylinder during the fuel combustion only a fraction is converted into useful work, the rest being wasted with the exhaust into the atmosphere and absorbed by the engine components in contact with the hot gases—products of combustion. Lack of cooling would not only cause a

nonuniform heating of these components. It would lead to their overheating above the permissible limit, giving rise to high thermal stresses. An overheating of the piston or cylinder liner leads to the evaporation and burning of lubricating oil with the deposition of lacquer and carbon which deprive the piston rings of elasticity and cause their sticking.

An inadequate cooling reduces the cylinder clearance and causes distortion and scuffing of the piston and cylinder liner. Thermal overstressing triggers cracks in the piston crowns which lead to combustion blow-by.

The function of the cooling system is to cool down the engine components, the lubricating oil and scavenging air to a point where optimum operating conditions are created. In shipboard propulsion plants, the cylinder heads, cylinder liners, exhaust manifolds and turbochargers are subjected to cooling with water. Water-cooled pistons is a feature of high-output crosshead diesel engines which compares favourably with the cooling by oil, assuring effective cooling at the piston rings. Telescopic pipes functioning independently of the rest of water lines are used to feed and drain the piston cooling water.

In use on some diesel engines is also a self-contained system for cooling the injectors. However, the cooling of pistons and injectors with water is possible only if the crankcase is reliably protected against the ingress of water. Otherwise, water may enter the crankcase and form there an emulsion with the oil which will interfere with normal lubrication and be detrimental for the engine.

Taking the heat liberated in an engine cylinder due to the combustion of fuel as 100%, the losses with the cooling water amount to 18-20% at the cylinder head, 10% through the piston and 5-8% through the exhaust manifold and turbocharger.

Trouble-free functioning of the cooling system is essential not only for normal operation of the diesel but also for the warming up of the engine before starting up, and is therefore a factor creating optimum conditions when starting and maneuvering a propulsion installation.

67. Cooling Systems of Modern Marine Diesel Engines

In a once-through or direct-flow cooling system (Fig. 88), pump *3* draws outside water through sea valve *1*, gate slide valve and filter *2*, delivering it into main water line via oil cooler *4*. Tappings *6* and *7* are taken to each cylinder of block *12* and to the water jacket of exhaust manifold *10*. Cylinder heads *11* are cooled by the water admitted thereinto from the upper part of cylinder block through pipes *9*, and the outflow from the heads is disposed of overboard

through line *8*. Valves provided in the line at each of the cylinders control the water outflow.

The water temperature at the outlet from each cylinder head must be maximum between 318 and 323 K (45-50°C), to avoid rapid scale formation in the water jacket. This temperature is monitored by thermometers fitted into each cylinder head.

Simple and convenient in operation as it is, the once-through cooling system fails to provide for uniform cooling of the engine,

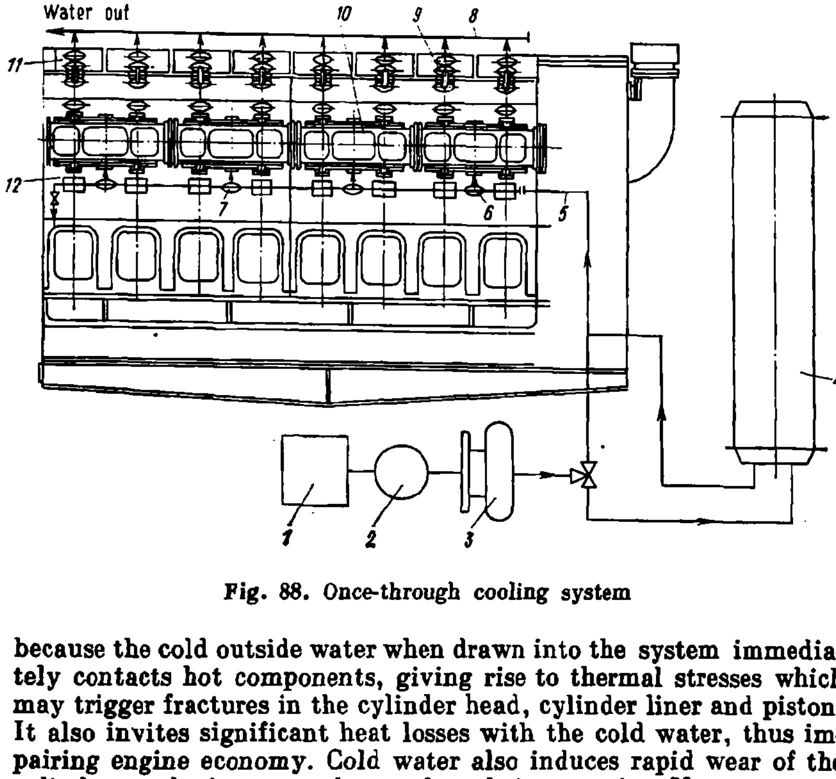


Fig. 88. Once-through cooling system

because the cold outside water when drawn into the system immediately contacts hot components, giving rise to thermal stresses which may trigger fractures in the cylinder head, cylinder liner and piston. It also invites significant heat losses with the cold water, thus impairing engine economy. Cold water also induces rapid wear of the cylinders and pistons, and provokes their corrosion. Numerous zinc corrosion inhibitors, which the cylinder block, cylinder heads and oil coolers are fitted with in an attempt to ensure normal functioning of the cooling system, may appear to be of no avail against a corrosive attack detrimental to these components. Therefore, the once-through system of cooling is no longer used on diesel engines of recent make.

Consider the layout of a closed-circuit cooling system (Fig. 89) made up, in fact, of two circuits: a closed circuit for fresh water intended to cool down the engine and an outside water circuit for

cooling the fresh water circulated through the engine as well as oil in special coolers.

Pump *15* feeds water over the internal circuit into cylinder block *1*. On cooling cylinder head *2*, water is taken via pipe *3* into water jacket *5* of the exhaust manifold and hence to thermostat (or a temperature controller) *8* which automatically controls the temperature of the fresh water circulated through the engine. The thermostat is arranged to by-pass the bulk of water into cooler *11* and the rest into pipe *7* when the jacket temperature is above the opening point

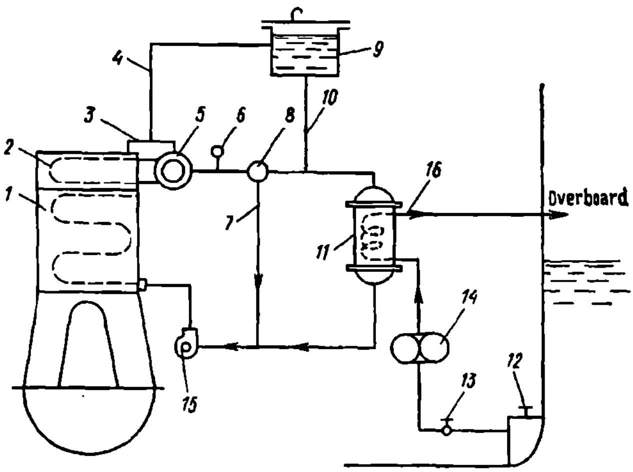


Fig. 89. Closed-circuit cooling system

of the thermostatic valve. Thus, the thermostat either directs the water into the pump *15* to cool the engine or diverts it into the cooler depending on the temperature monitored by thermometer *6*. The high temperature of the water outflow from the engine may cause the generation of some steam inside the water jacket of the engine which is delivered over line *4* into expansion tank *9* allowing for the expansion of the water on heating. Excess water is expelled from the tank into line *10*, thus preventing leakage through the joints of the cooling equipment. Outside water is drawn by pump *14* through sea valve *12* and intake valve *13* and is discharged overboard via pipe *16* on passing through water-by-water cooler *11*.

A closed-circuit cooling system provides for an outflow temperature of 353-363 K (80-90°C) against an inflow temperature of 323-333 K (50-60°C) depending on the size and type of the engine, thus creating conditions for trouble-free operation. In low-speed engines, the temperature difference between the water outflow and inflow

is commonly maintained at 7-9°C. An increase in the jacket water temperature shortens the ignition delay with the result that the rate of combustion pressure builds up and the final pressure decreases to a point where the fuel burns up in a gradual way, thus enabling the engine to run in a smooth way. It will be noted that in cooling systems connected to the atmosphere the jacket water temperature must be below the boiling point. This requirement is of particular importance for high-speed diesel engines which reject to the cooling water more heat than this is the case for low-speed engines.

The smaller the size of a diesel engine cylinder, the greater is its relative cooling surface area (the ratio of the cylinder surface area and swept volume), for areas and volumes are proportional to the linear dimensions raised to the second and third power, respectively. Let us explain this by way of an example. If there are two single-cylinder engines with the cylinder bores of 100 and 300 mm and the piston strokes of 120 and 360 mm, respectively, the cooling surface area of one engine is

$$2 \times 3.14/4 \times 10^2 + 3.14 \times 10 \times 12 = 534 \text{ cm}^2$$

and the swept volume is

$$3.14/4 \times 10^2 \times 12 = 940 \text{ cm}^3$$

Correspondingly, the cooling surface area of the other engine is

$$2 \times 3.14/4 \times 30^2 + 3.14 \times 30 \times 36 = 4\,800 \text{ cm}^2$$

and the swept volume is

$$3.14/4 \times 30^2 \times 36 = 25\,400 \text{ cm}^3$$

Thus, the relative cooling surface area of the former engine is $534/940 = 0.57$ cm^{-1} and that of the latter is $4\,800/25\,400 = 0.19$ cm^{-1} or only 1/3 of the smaller engine. This implies that the smaller the engine the more heat it rejects to the cooling medium. Consequently, in smaller diesel engines it is necessary to maintain a higher jacket water temperature than in larger ones, at least 323 K (50°C), in order to obtain an adequately high compression temperature. Particularly important is the requirement when the engine should run at low idling speed on starting from cold, because a low water jacket temperature leads to incomplete combustion and, as a result, to the gumming of pistons, piston rings and valves.

The closed-circuit cooling system prevents the scaling and thermal overstressing of the engine. Practical experience goes to show that it also reduces wear in the engine and materially speeds up its warming up in cold starts.

The system is provided with high pressure and high temperature alarms of the visible and audible type operated with the aid of pres-

sure- and temperature-sensitive relays mounted in the fresh water circuit.

A standby water pump is incorporated into the system, to take over in the event of a failure of the main electric pump. The fresh water circuit is fitted with a heater used for warming up the water and engine to at least 333 K (60°C) after the engine has stood idle for a considerable period of time.

The so-called high-temperature cooling, i.e. one effected at a jacket water temperature over 373 K (100°C), is an innovation aimed to efficiently recover the heat of cooling water and exhaust gases and to increase the efficiency by 5-7%, i.e. to 0.45-0.47. Since the jacket water temperature exceeds the boiling point, the steam thus generated to some extent is tapped to meet various shipboard needs. However, this is not the only advantage of high-temperature cooling. For example, admission of a mixture of water and steam into the water cooler of the engine makes it possible to cut the active surface of the cooler to a considerable extent, rendering it a low-weight unit, and to save the non-ferrous metal used for its manufacture. Apart from that, no rapid wear inherent on engines digesting high-sulphur fuel is noted if the cooling system is of the high-temperature type. An increase in the cylinder wall temperature calls, however, for the use of high-quality compounded oil.

68. Water Pumps

To feed water into the cooling system, diesel engines employ reciprocating or centrifugal pumps, alongside with self-priming liquid-packed ring pumps.

Reciprocating water pumps are commonly a feature of low- and medium-speed diesel engines of small power output. They are actuated either by the scavenging pump, using the so-called balance beam, or by a crankpin directly. Capable of creating a good suction lift, these pumps can also operate against a significant delivery head, displaying high efficiency and feeding water in one direction only irrespective of the crankshaft rotation direction.

Let us look at the design features of the reciprocating water pump used on the diesel engines of the ДР 30/50 range.

Piston *13* (Fig. 90) provided with rubber packing rings and actuated by drive *9* and *10* linked to the scavenging pump crosshead reciprocates in a bronze sleeve press-fitted into cast iron casing *14*. The casing is provided with two spaces contained in which are valve chests *2* and *4* each accommodating six interchangeable rubber valves. The lower end face of the piston interacts with three suction valves *3* of lower valve chest and three delivery valves *8* of the upper valve chest, the two chests being arranged in the right-hand spaces of the casing and communicating with the piston via passages *11*

and *12*. The upper end face of the piston interacts with the rest of the suction and delivery valves in the left-hand spaces.

Suction produced by the piston on a downstroke opens inlet valves *3* in the corresponding valve chest through which water is admitted into the space above the piston. At the same time, a pressure buildup below the piston opens delivery valves *8* through which water enters pipe *7* of the cooling system, a wall not shown in Fig. 90

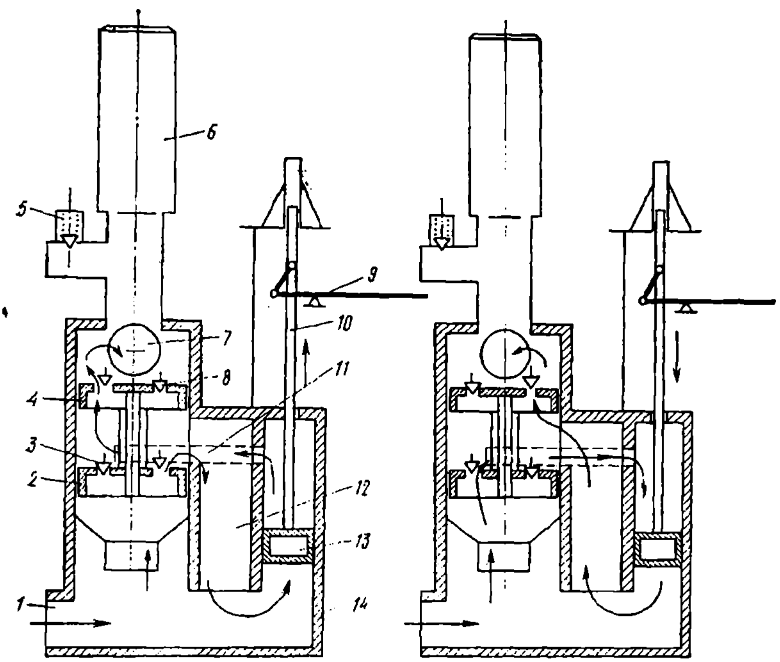


Fig. 90. Reciprocating water pump

separating spaces *1* and *12* from each other. The operation is repeated in the reverse order when the piston is displaced towards TDC.

To take up irregularities and induce a uniform flow, cushion chamber *6* is fitted to the casing. The air compressed in the cushion chamber during the delivery stroke of the piston, expands in the course of the suction stroke, maintaining a constant pressure at the delivery side of the pump. Relief valve *5* is commonly set to open at a pressure of 29.4×10^4 N/m^2 (3 kgf/cm^2). Pump delivery is adjusted by changing the effective piston stroke. The stuffing box prevents water leaks along the piston rod. The pump can be drained from water through a cock provided in the casing.

Reciprocating pumps have a number of drawbacks such as irregular flow of water during the delivery stroke, complicated drive and construction (applies to the suction and delivery valves), large size and mass, limited service life and low capacity.

Centrifugal pumps are free from these drawbacks and are therefore widely used in marine practice. They are compact, balanced and low-weight units characterized by high efficiency and trouble-free,

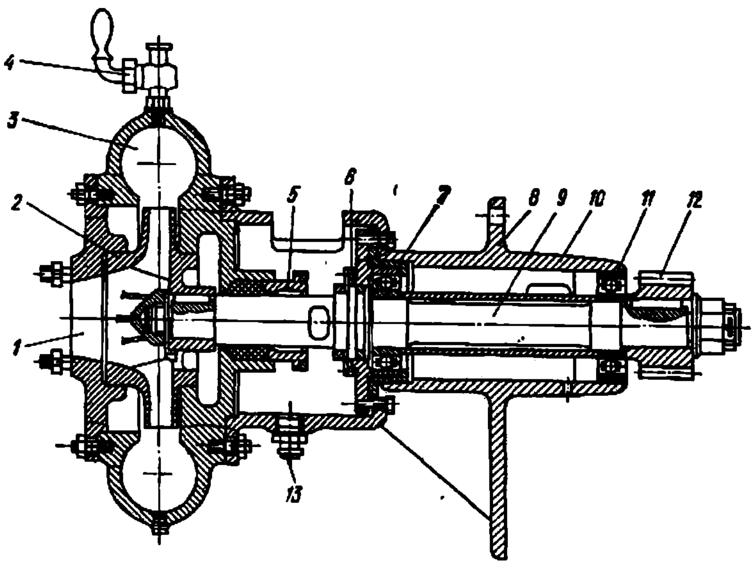


Fig. 91. Centrifugal pump

noiseless operation at high speeds ensuring uniform water flow. A low suction lift, commonly between 2.94×10^4 and 4.9×10^4 N/m^2 (3-5 mm Hg), is their only limitation.

A centrifugal pump may be driven by an electric motor or from the crankshaft through gear *12* (Fig. 91). If crankshaft-driven, the pump is attached to the engine bedplate with the aid of bracket *8* cast integrally with casing *10*. A suction cover at the front end is provided with flange *1* connected to which is a suction line. The inside of the casing is a water-way termed volute *3*, which increases in cross section and becomes a discharge bowl to pass the entire water flow before it enters the delivery line. The rotating element consists of impeller *2* with vanes and shaft *9*, the former being keyed to the latter. The shaft is supported by ball bearings *7* and *11* protected against the ingress of water by cover *6*. Stuffing box *5* consisting of an impregnated cotton packing held fast by a flanged gland prevents water leaks along the pump shaft.

In operation, the water entering the impeller is discharged radially into the volute due to the centrifugal forces, reaching then the delivery line. At the same time a vacuum, created in the central part of the impeller due to the displacement of the water towards its periphery, provides for a sucking action in the line.

To ensure trouble-free operation, centrifugal pumps need to be primed preparatory to starting the engine. Before a long interruption of service, water must be drained from the pump through pipe *13* and air must be vented through cock *4* fitted to the casing at its topmost point.

69. Water Coolers and Water Temperature Controllers

Fresh water cooling is effected in a water-by-water cooler employing the outboard water as a cooling medium.

Water-by-water coolers are either of plate or of shell-and-tube type (with flat tubes), and are similar in their design to oil coolers. The former are more efficient than the latter but less popular in marine application owing to the size and mass posing installation problems in the cramped engine room.

The common type used on board ships is the shell-and-tube water cooler. Fresh water is passed through the tube bundle cooled by the outside water circulated through the shell. Provided in the shell baffle plates lengthen the path of the outside water.

Engine output, wear on the cylinders and pistons, and specific fuel consumption are all influenced by the jacket water temperature. Therefore, it must be maintained at a level ensuring maximum output, and minimum wear and fuel consumption. Temperature controllers are incorporated in the cooling system to cater for optimum temperature conditions under various engine loads. Depending on the mode of operation, controllers are of the indirect- and direct-action type. Whatever the type, the main requirement for automatic temperature control to be met is a trouble-free service.

A temperature controller (Fig. 92) incorporates transducer *1*, bellows *2*, set-point device *3* and final control element *4*. The transducer, which may be of the liquid-filled or expansion type, responds to temperature changes in the cooling system.

A liquid-filled transducer consists of a sealed filled system containing a liquid which may be ethyl alcohol, ethyl ether, acetone and the like. When an increase in temperature causes the liquid to evaporate, the vapour exerts pressure on the final control element, and activates it.

An expansion-type transducer comprises a tube and a rod made of materials having different coefficients of linear expansion. Invar, the material of the rod is an alloy containing 64% of iron and 36% of nickel, and has the coefficient significantly smaller than that of

the tube's material. A change in the temperature of the medium causes a proportional displacement of the rod.

Being valves of various kinds, final control elements serve to divert the flow of fresh and outside water from a cooler into a by-pass and vice versa, depending on the temperature.

Indirect temperature controllers are widely used on marine diesel engines with an output over 1 470 kW (2 000 hp). One depicted in

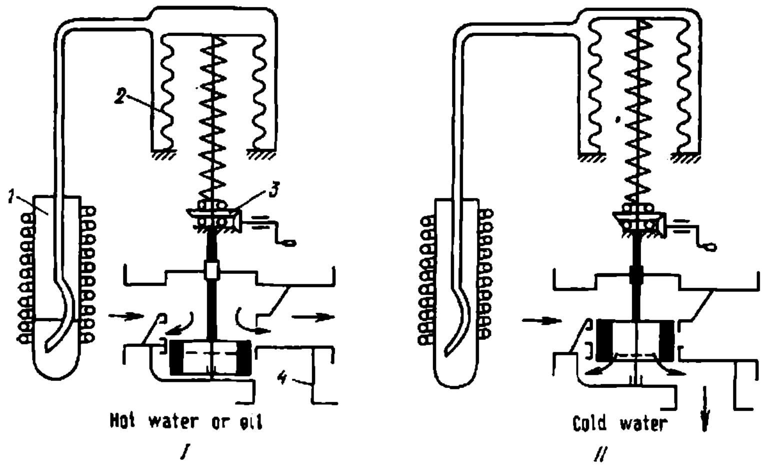
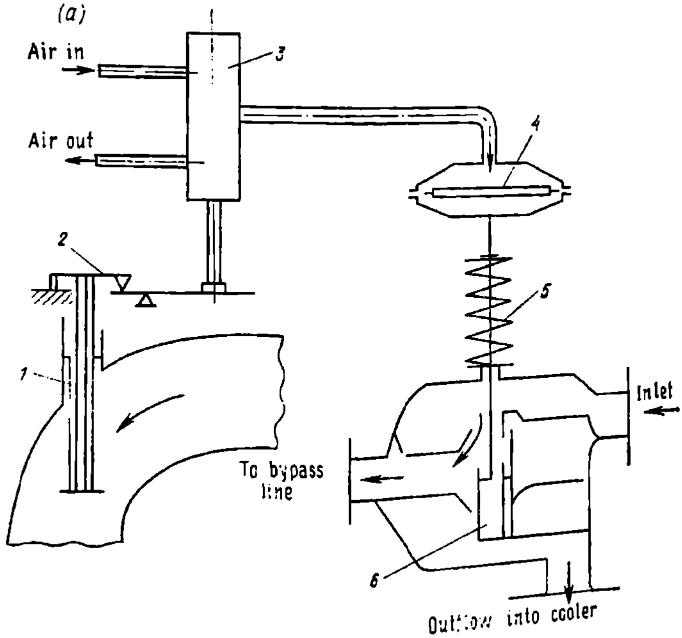


Fig. 92. Positions assumed by final control element of temperature controller in response to hot (*I*) and cold (*II*) water flow

Fig. 93*a* consists of pneumatic control unit *3* with transducer *1* and set-point device *2*. Responding to a change in temperature, the transducer of the expansion type either expands or contracts, its motion being transmitted to final control element *6* by way of a leaverage. A control valve inside pneumatic unit *3* either controls the flow of compressed air from a bottle into the space above the diaphragm of actuator *4* of the final control element or discharges air into the atmosphere, relieving the line of the pressure. A change in the operating air pressure causes actuator *4* to displace final control element *6* against the action of spring *5* with the result that the amount of water entering the cooler or by-passing it is also changed. A given set point is adjusted with the aid of a handle (see Fig. 92) which, in its turn, changes the tension of the control valve spring. The temperature setting is indicated by a pointer and a scale. The handle is also used to control the temperature if the pneumatic actuator becomes inoperative.

The disadvantage of indirect temperature controllers is a high consumption of compressed air.

In a direct-action temperature controller known as thermostat (Fig. 93*b*), transducer *2* is contained in a case and linked directly



to the final control element in the form of two valves *1* and *3* termed the main and by-pass valve, respectively. The transducer consists of two spring-loaded bellows interconnected by a bracket. The bellows springs are adjustable for tension with the aid of screws. The bellows are filled with a volatile liquid. Its volatilization due to a rise in the temperature of the surrounding working fluid causes the bellows to expand so that the jacket water is directed into the cooler.

Direct-action temperature controllers find application in diesel engines with an output less than 1 470 kW (2 000 hp). In case of the bellows developing a leak,

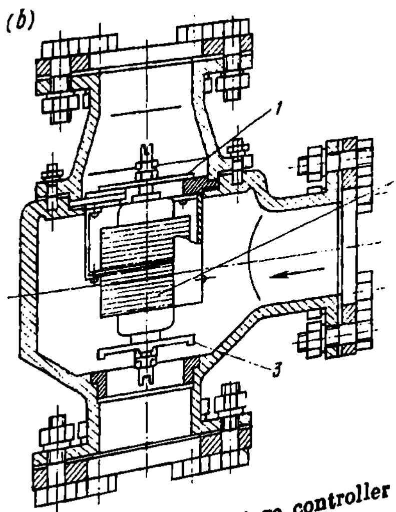


Fig. 93. Temperature controller

these controllers are arranged to direct all the jacket water through the cooler. Indirect temperature controllers are free from that shortcoming.

Lubricating systems are also fitted with temperature controllers by analogy with cooling systems.

70. Cooling System Malfunctions

The use of outside water or one which has not been appropriately tested and accordingly treated inevitably leads to the scaling and corrosion of the metal surfaces in contact therewith. The scaling interferes with the rejection of heat through the cooling surfaces, causes local overheating and thermal overstressing of parts and is likely to trigger cracks. Descaling carried out in good time provides for normal operation of a diesel engine.

Practical experience goes to show that in a closed-circuit cooling system high temperature may result from a defective pump, a clogged filter or cooler. The remedy is to check the pump for operation and recondition it, if necessary, or to clean and rinse the defective filter or cooler.

A water pump may cease or reduce delivery due to a heavy inleakage of air through a suction pipe joint, or the stuffing box, a defective valve or bad wear on the packing rings (applies to reciprocating pumps). The defect is to be taken care of by inspecting all the joints of the suction line, taking apart and repacking the stuffing box, disassembling the valve chest, renewing the defective valve, replacing the worn packing rings by new ones.

A destructive corrosive attack on elements of the cooling system results from the oxidation of the metal by the oxygen contained in water. Most of the phenomena involving the corrosion of metals are electrochemical in nature. The surfaces in contact with outside water are protected against destruction by zinc corrosion inhibitors fitted inside the cylinder water jacket, cylinder head and water cooler. In recirculating fresh water cooling systems protection is given by treating the water with chemicals forming oxide films on the heat-exchange surfaces. Also in use are slushing oils which produce coatings strongly adhesive on metals without interfering with heat transfer. Procedures for any of such additives utilization are outlined in relevant instruction manuals.

REVIEW QUESTIONS

1. What purpose does the cooling system of a diesel engine serve?
2. How is the cooling of pistons effected?
3. What cooling systems are used on marine diesel engines?
4. What is the function of water pumps?
5. Describe the construction of reciprocating and centrifugal pumps.

6. What is the essence of the high-temperature cooling system?
7. What purpose do water coolers serve?
8. What is the function of temperature controllers? Discuss their construction and operating principle.
9. What are the main malfunctions of the cooling system and methods of their elimination?

Chapter XII

THE COMPRESSED AIR SYSTEM

71. Shipboard Sources of Compressed Air

Medium- and high-output diesel engines are started and reversed with the aid of compressed air supplied from air cylinders under a pressure of 294×10^4 N/m^2 (30 kgf/cm^2). High-speed diesel engines require a higher starting air pressure.

Shipboard air cylinders vary in capacity depending on the size of the engine, its power and application. The USSR Register stipulates that starting air is to be kept in stock on board a ship in two cylinders of equal capacity sufficient for twelve or six cold starts of direct-reversing or unidirectional engines respectively. Air cylinders are to be charged each one to a pressure of 294×10^4 N/m^2 (30 kgf/cm^2) by means of a compressor maximum during one hour. However, since most of the commercially-available compressors with individual drives develop a pressure of 588×10^4 N/m^2 (60 kgf/cm^2), marine air cylinders are also rated for the same pressure, this practice effects savings in weight and space. To obtain a supply of starting air at 294×10^4 N/m^2 (30 kgf/cm^2), a pressure reducing valve is introduced between the air cylinders and engine.

The shipboard air compressors in use are electrically-driven machines of the two- and four-stage type with intercoolers between the stages, equipped with plain or differential pistons. The single-stage compressor depicted in Fig. 94 is designed for a maximum pressure of 38×10^4 N/m^2 (4 kgf/cm^2). The outflow of compressed air is passed through an oil trap. Intercoolers of the shell-and-tube or coil type are provided between the stages of multistage compressors to cool the air by outside water.

Compressor capacities range between 25 and 500 m^3/h. So, for example, the standard equipment of the ДКРН 55/110 diesel engine made at the Bryansk Engineering Works is two 65-m^3/h compressors charging two 3-m^3 air cylinders, the two 11-m^3/h air cylinders of the 8ДКРН 74/160 diesel are charged with the aid of two 282-m^3/h

compressors; and the two 18-m^3 air cylinders of the Sulzer 9RND-90 model are served by two compressors, each rated for 500 m^3/h.

Although the engine-starting service commonly requires 294×10^4 N/m^2 (30 kgf/cm^2), the actual pressure capable of setting an

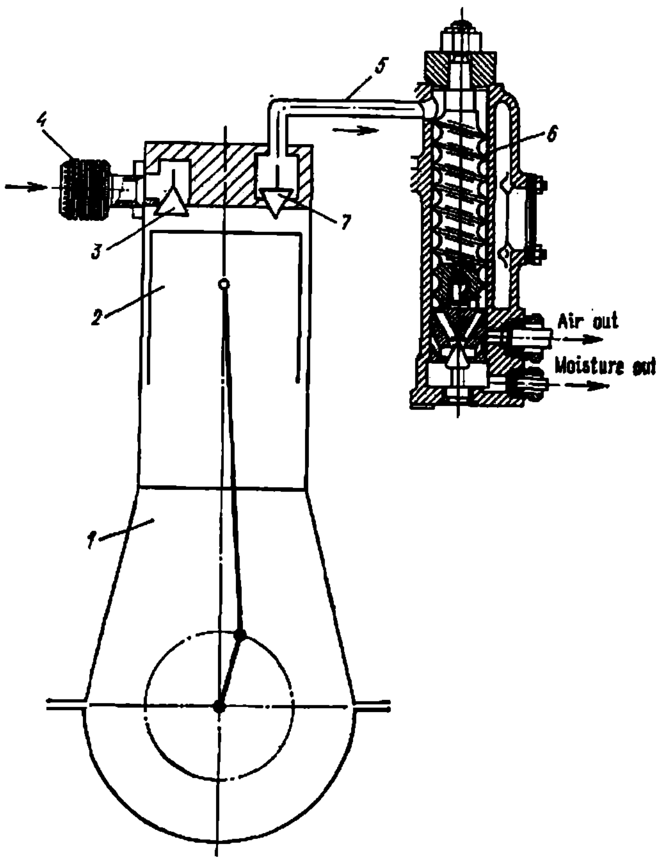


Fig. 94. Single-stage air compressor

1—compressor frame; 2—piston; 3—suction valve; 4—suction line; 5—delivery line; 6—moisture-and-oil trap; 7—delivery valve

engine into operation depends on its temperature. For example, a warmed-up 6ДР 30/50 diesel will pick up the cycle under only 78×10^4 N/m^2 (8 kgf/cm^2).

High-speed diesel engines operating at 1 500-1 700 rpm call for higher starting air pressure, because their cranking speed is 130-150 rpm as compared to 40-60 rpm in low-speed large-bore units.

This difference is due to the fact that the heat losses on smaller engines are higher than on larger ones because of the relative cooling surface of the former being greater than that of the latter. Consequently, a small engine is to be cranked at a high speed to achieve compression temperature producing ignition in spite of the heat losses incurred. Particularly high are heat losses in the high-speed diesel engines whose cylinder heads and pistons are made of aluminium alloys.

72. Air Cylinders

Marine propulsion installations are provided with compressed air cylinders of a capacity varying with the swept volume of engine cylinders, the specified number of cold startings (6 or 12)

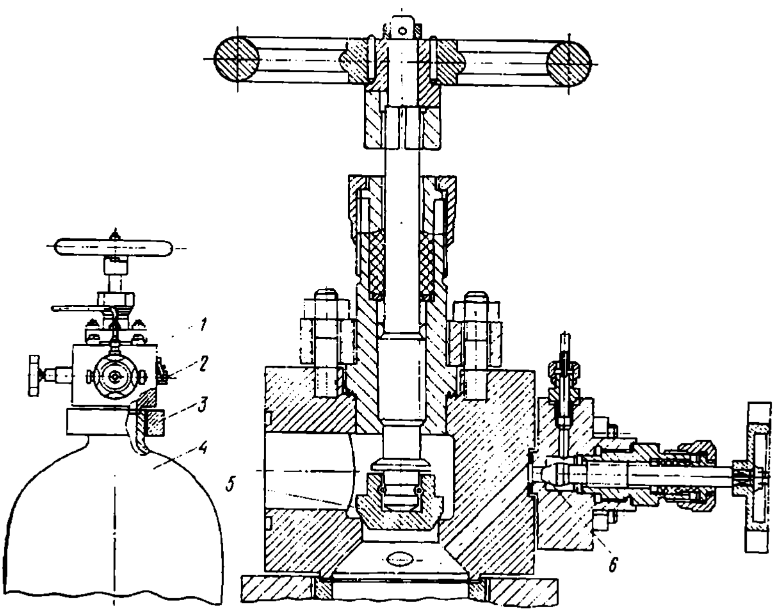


Fig. 95. Starting air cylinder

and the air requirement per starting operation. Commonly, this requirement is 10-12 l per litre of the swept volume in starting an engine from cold, and 5-8 l for starting a warmed up engine. The starting facility of a Fiat 909S diesel engine includes two 9-m^3 air cylinders, that of a MAN K7Z 84/160 diesel incorporates two 11-m^3 air cylinders; a Sulzer 9RND-90 diesel starting facility has two

18-m^3 air cylinders and that of a 12K84EF diesel engine is two 26-m^3 air cylinders.

Referring to Fig. 95, air cylinder *4* is a seamless or welded steel vessel having flange *3* with head *1* superposed by various fittings such as shut-off valve *5*, feed valve *6* and safety plug *2*. The shut-off valve disconnects the air cylinder from the line, the feed valve taps air to the whistle and for domestic needs, and the safety plug of the fusible type bleeds air from the cylinder in the event of an excessively high engine room temperature.

In operation, steps are to be taken to prevent accumulations of condensate or oil in the air cylinders. The condensate formed due to the condensation of water vapour contained in the air may cause corrosion of the engine cylinders and interfere with normal starting and reversing of the engine. Starting air contaminated with oil may explode anywhere in the air line between the starting air master valve and starting valves of the engine cylinders.

The operating instructions stipulate that each air cylinder should be registered and certified with an appropriate state technical supervising authority. The air cylinders are to be tested hydrostatically every six years with the authority's representative in attendance. The results of the test are entered into the certificate (serial No. of the air cylinder, test date, the pressure the air cylinder is rated for, the date of the next hydraulic test, data about the fittings, etc.).

73. Starting Air System Malfunctions

The nature of wear on compressor components is identical with that on other engine components such as piston rings, bearings, cylinders and crankshafts. Defective compressor components are reconditioned in the same way as the engine parts. The plate valves of a compressor are lapped on a plate and seated. Annealed copper gaskets are applied under the valve bodies for air tightness.

After a period in service an explosive sludge accumulates on the delivery valves, inside the cylinders and coil tubes of the intercoolers. Therefore, the regular cleaning of sludge deposits is essential. The same applies to the descaling of intercooler tubing which impairs compressor performance.

Relief valves are to be inspected and adjusted at regular intervals. A seal is to be affixed to the checked valve. Rupture discs showing traces of deformation should be renewed.

The air-distribution valve is less susceptible to wear, because its operation is limited to the engine starting only. The seizure of spool valves or rotor is eliminated by lapping these parts in the valve body.

Major defects occurring in starting air cylinders are wear on fittings

and sludge deposits inside the cylinders. For trouble-free service, the air cylinders shall be cleaned at regular intervals and the fittings inspected and lapped, if necessary.

REVIEW QUESTIONS

1. How is compressed air obtained and used on board a ship?
2. What is the construction of a compressor?
3. Why is it necessary to cool the air in a compressor?
4. Why is it necessary to clean the air?
5. What are the main components of an air cylinder?
6. What are the factors that affect the air consumption at engine starts?
7. What are the main malfunctions of the compressed air system and how they can be eliminated?

Chapter XIII

THE STARTING AND REVERSING GEAR

74. Conditions for Starting a Diesel Engine

The starting of diesel engine is one of the crucial operations dependent upon which is normal service of the engine. A reliable starting is also a factor that materially influences the ship's maneuverability. The starting is effected by setting the crankshaft to rotate at a speed that will produce the ignition temperature in the engine cylinders. The starting falls into three stages which are the cranking of the engine by means of compressed air or any other starting method until some of the cylinders begin to fire, picking up the cycle on fuel without the engine's misfires and acceleration to a speed in accordance with the fuel-injection pump setting and the warming up of the engine with a gradual increase in load to a given value.

Marine practice favours the starting with compressed air. High-speed diesel engines with an output less than 589-736 kW (800-1 000 hp) may employ the starting by either air or electricity. Whatever the method in use, the main requirement is that the crankshaft should be rotated at minimum cranking speed, n_{min}; commonly, $n_{min} = 0.12n_r$-$0.15n_r$. To effect the starting by air irrespectively of the position of the crankshaft, a two-stroke engine must have at least four cylinders and a four-stroke engine, six cylinders. Each of the engine cylinders, whatever their total number, must be fitted with a starting, air line of its own.

To obtain a trouble-free starting, the cranking period may vary with the condition of air-fuel intermixture, fuel self-ignition, and

the state of the engine. The USSR Register rules call for the builder's guarantee that the engine will start-up from cold failure-free at an engine room temperature of at least 8°C. The period which elapses before the engine is under own power on being cranked by air is between 2 and 8 s. At this stage, the running is irregular and the exhaust is smoky.

The irregular running is due to the fact that some of the cylinders misfire and the engine gains speed in jerks as the cylinders pick up the cycle on fuel one after another.

The starting air gear of a direct-reversing diesel engine comprises a starting air master valve, starting valves in the cylinders, an air distribution valve, an interlocking gear, pilot valves, air cylinders and an air line with a shut-off valve before the engine.

75. The Starting Air Master Valve

The starting air master valve provides a quick connection of the starting air line to the air cylinders and disconnects the cylinders as soon as the engine picks up the cycle on fuel. The feeding of air to various items of the starting and reversing gear and the bleeding of compressed air from the line are effected automatically depending on the position of the starting and reversing gear, and of the fuel equipment.

Starting air master valves are classed with reference to the operating principle into two types: unbalanced and balanced valves. Coming under the former heading are the valves which are opened due to relieving the pilot piston of an air pressure, and the latter heading include those opening under an air pressure applied to the pilot piston.

A starting air master valve of the unbalanced kind illustrated in Fig. 96*a* moves in sleeve *2* due to the action of spring *1*. Collar *8* provided between the external and internal parts of valve *3* serves to open it in response to a differential pressure created when air is admitted through metered opening *4*. At starting, fed into a space *b* through this opening some of the air reaches a pilot valve on the engine room control console through pipe *10*. The setting of the starting lever to the "START" position opens the pilot valve which relieves the internal space of pressure. The resulted differential pressure displaces valve *3* upwards clearing the way for the air into pipe *7* and hence to the starting valves of the engine. On completion of the starting operation when the starting lever is returned to its original position, the pilot valve is automatically closed so that the internal space of the master valve becomes disconnected from the atmosphere and the valve itself is closed due to a combined action of compressed air and spring *1*. The remaining starting air is vented from the air line into pipe *6* by way of a flat on valve stem *5* at the

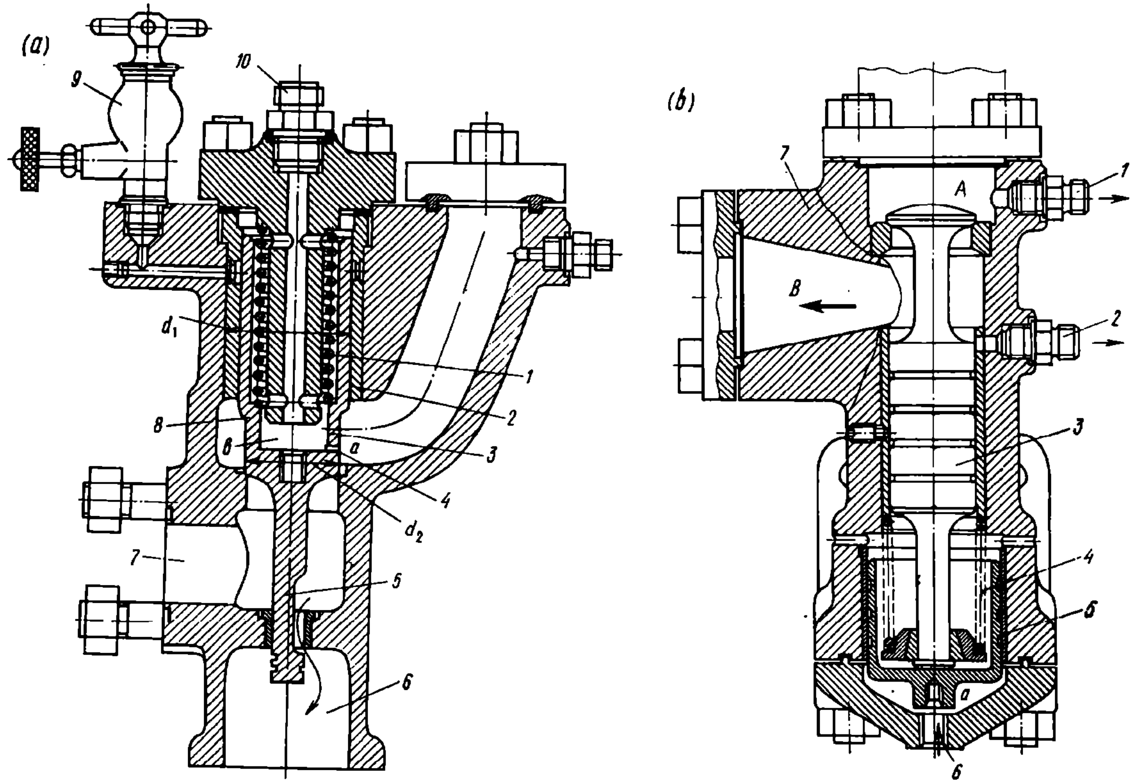


Fig. 96. Starting air master valves

instant the valve settles down. Pipe *6* is connected to either the air receiver or exhaust manifold. The valve is lubricated with the aid of grease cup *9*.

The possibility of seizing due to carbon or scale entering the starting air line is a limitation of the unbalanced master valve.

A balanced master valve depicted in Fig. 96*b* is more reliable in service. It consists of valve *3* proper contained in valve body *7* and provided with pilot piston *5*. Valve *3* is held closed with spring *4* on the one hand and with the pressure of air on the other hand. Some of the air admitted into the master valve is tapped to a pilot valve on the engine room control console via pipe *1*.

To open the starting air master valve, the starting lever is shifted from the "STOP" to the "START" position. The pilot valve opens, and the air enters the space *a* in the pilot piston via pipe *6*. The difference between the pressure applied to the pilot piston and that acting on a disc causes valve *3* to open. As soon as the engine is under own power, the pilot valve automatically closes and so does the master valve, being acted upon by spring *4* and relieved of the differential pressure in the upper space and the space *a*. After that, the air is vented into the air receiver or exhaust manifold via pipe *2*.

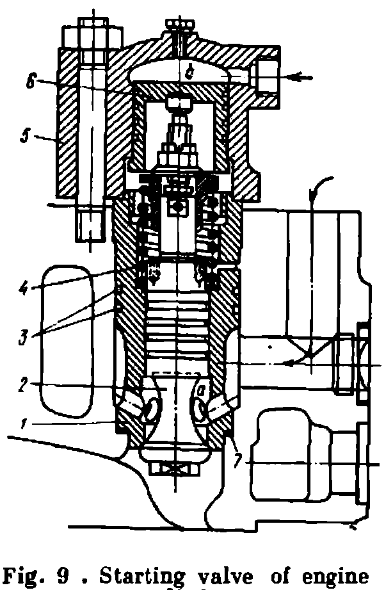


Fig. 9 . Starting valve of engine cylinder

76. Starting Valves of Engine Cylinders

The starting valve of an engine cylinder may be thought of as an actuator of the starting and reversing gear. By analogy with the starting air master valves, the starting valves may be classed depending on the mode of operation into balanced and unbalanced types.

Balanced starting valves are in frequent use. Valve *2* (Fig. 97) is contained in body *1* with cover plate *5* overlaying pilot piston *6*. Spring *4* holds the valve in its seat.

In starting the engine, when the master valve admits air into the line, the space between the valve body and the lower part of the starting va.ve, each of the engine cylinders is fitted with, becomes filled with air. However, the starting valve can open only then when the air fed from the distribution valve enters the space above the pilot piston and the pressure thus applied to the guiding portion

of the stem by way of the pilot piston exceeds the total pressure sustained by the lower part of valve *2*. Once the starting air has been admitted into the engine cylinder and the crank has turned through a specified angle, the starting air distribution valve connects the space above the pilot piston to the air receiver, relieving the piston of the pressure and enabling spring *4* to close valve *2*. A sealing arrangement consisting of copper gasket *7* and rubber rings *3* provided at the valve base and in the air space, respectively, serves to prevent compression blow-by between the cylinder head and starting air body.

77. The Starting Air Distribution Valve

The starting air distribution valve serves to control the functioning of the starting valves in accordance with the firing order.

Shipboard practice knows starting air distribution valves of two types, *viz.* spool valves and rotary valves. The number of spool valves in an air distribution valve of this kind depends on the number of the starting valves. The spool valves are actuated either by special cams on the camshaft, or by a separate drive, and are returned to the original position by springs. They are in limited use, being susceptible to the corrosive effect of the water vapour contained in air which leads to seizing.

Rotary starting air distribution valves are free from the defect of spool valves and are employed on a large scale. Consider the construction of the rotary distribution valve (Fig. 98*a*) used in the Д and ДР 30/50 diesel engines. It consists of body *4* containing cylindrical distribution block *5* with vertical drillings *10* in its upper end face and rotor *2* fitted to vertical shaft *7* driven from the camshaft through bevel gear *8*. Distribution block *5* has as many passages *11* in its side surface as there are engine cylinders, these passages being connected each to corresponding drillings *10*. Rotor *2* is provided with a passage of a diameter equal to that of the vertical drillings *10*. On opening the starting air master valve, air enters the space between rotor *2* and cover *1* of the distribution valve, pressing the rotor against distribution block *5*. Since at this stage the passage in the rotor registers with drillings *10* connected to the starting valve of an engine cylinder which is in a position ready for cranking, air continues its travel into this cylinder. Thus, the rotor, actuated from the camshaft, becomes connected to the rest of the drillings in the distribution block which, in their turn, are connected by lines to the starting valves in a way prescribed by the firing order. The successive positions of the rotor are shown in Fig. 98*b*.

While the rotor is a sliding fit on a key, permitting vertical displacement along shaft *7*, the distribution block is arranged to turn about the same shaft from one extreme setting into another, cor-

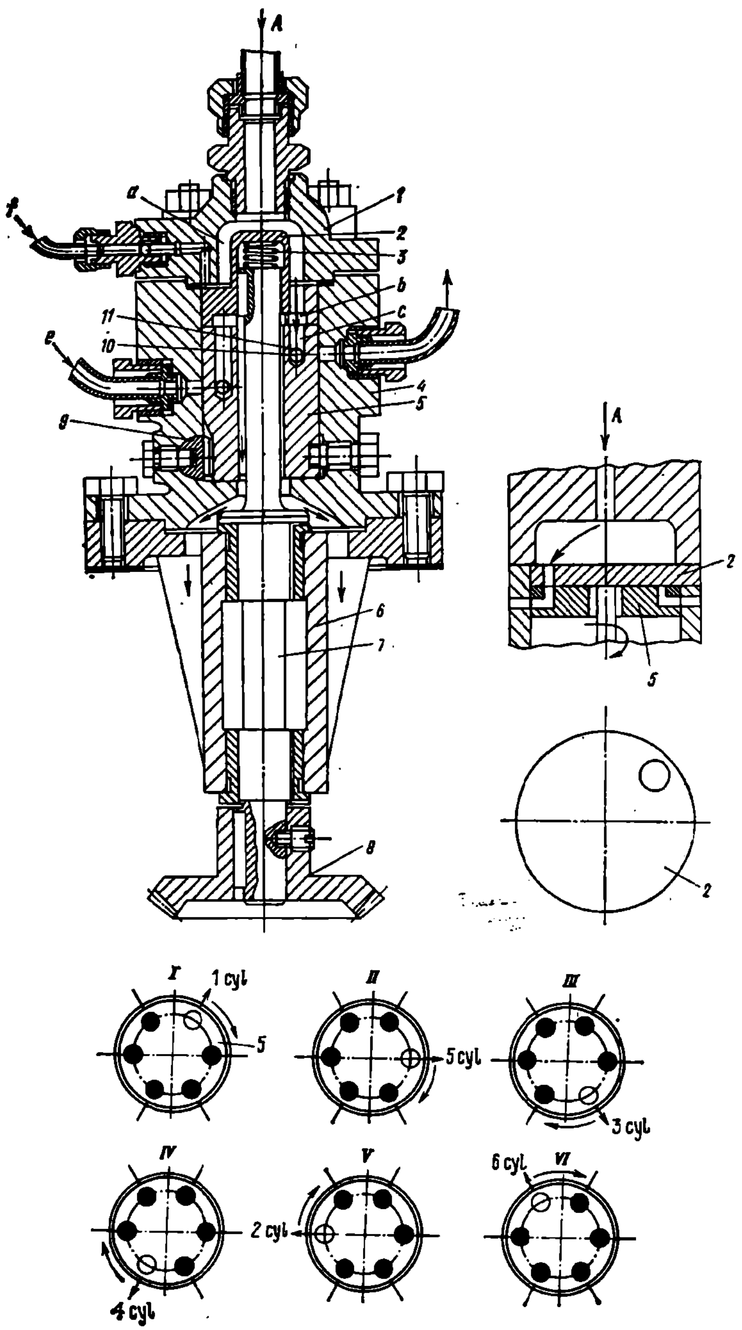


Fig. 98. Starting air distribution valve with cycle diagram

responding either to the right-hand or to left-hand rotation of the crankshaft at starting. Thus, the distribution block provides for a firing order in running ahead and one for running astern depending on the setting which can be changed with the aid of rack *9* meshing gears on a side surface of the distribution block and linked to the reversing handle by way of a tie rod.

On the delivery of starting air into one cylinder, its starting valve closes, and the process of feeding starting air into the rest of the engine cylinders is repeated cylinder by cylinder in accordance with the firing order.

When the engine has picked up the cycle on fuel, the rotor goes on rotating because of being rigidly linked to the camshaft. To minimize wear on the end face of the rotor, it is set out of contact with the distribution block by spring *3*. At the same time, the drilling of the block registering with the opening in the rotor is connected to a relief valve to relieve the starting valve of pressure.

Flanged bracket *6* serves to mount the starting air distribution valve on the engine.

78. Electrical Starting

High-speed internal-combustion engines with a power output under 735 kW (1 000 hp) can be started by an electric starter motor. It is a compact direct-current unit having a starter pinion fitted to the shaft and engaging the flywheel gear ring. Operated by a storage battery, the starter motor cranks the engine until it is accelerated to the cranking speed and begins to fire. After that, the starting pinion is pulled out of mesh with the flywheel gear ring by a special device.

Starter motors are rated to operate from 12-, 24- and 32-volt supplies and are capable of producing a high torque for a period of 10 to 15 seconds. To that end, their field and armature windings consist of a few turns of heavy wire having a low resistance. When a starter motor is energized, it may draw a current of 600-900 A. For accelerating the engine to the cranking speed, in use are various arrangements. For example, in wide-spread application is the so-called Bendix drive consisting of helical splines surrounded by the starter pinion. Also employed are solenoids which slide a shaft and the starter pinion in engagement with the flywheel ring gear.

The starting system depicted in Fig. 99 incorporates four-pole series- or compound-wound direct-current motor *1*. For setting starter pinion *6* in engagement with gear ring *7* of the flywheel and energizing the starter motor, use is made of solenoid *4* energized with the aid of control relay *3* which closes the contact points. The solenoid has a potential coil and a series coil producing a magnetic field which pulls in the core fitted with a contact disc at one end

and with pinion-actuating lever *5* at the opposite end. The disc closes the circuit of the starter motor and storage battery *2*, energizing the motor. At the same time, the contact disc of relay *4* cuts out the potential coil so that the core becomes retained by the series coil

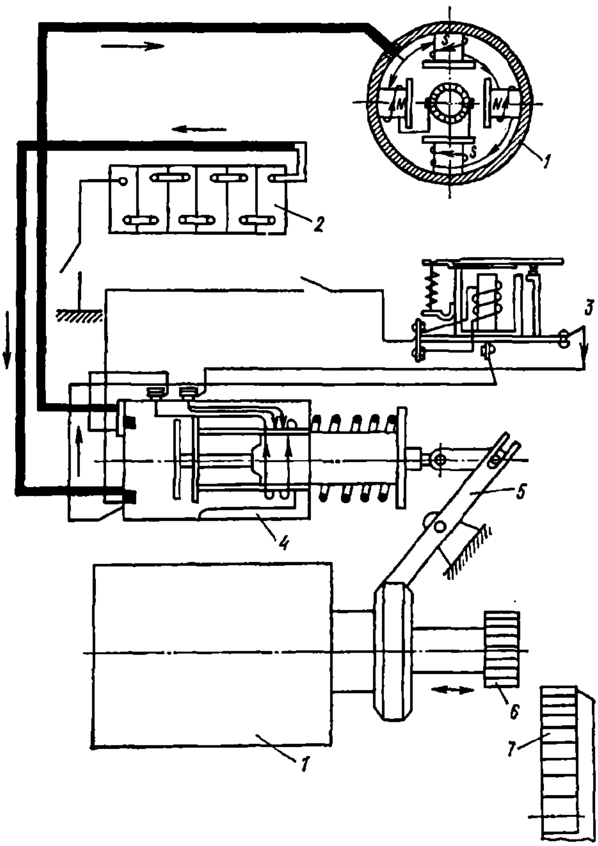


Fig. 99. Schematic of electric starting system

only. The displacement of lever *5* brings the starter pinion *6* into engagement with gear ring *7* of the flywheel. When the engine starts firing and the ignition key (applies to gasoline engines) or the starting pushbutton (on a diesel engine) changes the position, the control relay becomes de-energized and the circuit of the potential and series coils opens. The magnetic flux ceases to exist, and a coil spring pulls out the solenoid core and disengages the pinion from the ring gear.

79. Reversing Gears of Two- and Four-Stroke Engines

Two-stroke diesel engines use various reversing arrangements. The most simple one consists of a cylindrical distribution block turnable inside the starting air distribution valve and of symmetrical fuel cams. This method of reversing finds application on those two-stroke engines which require no altering of the valve timing in running astern.

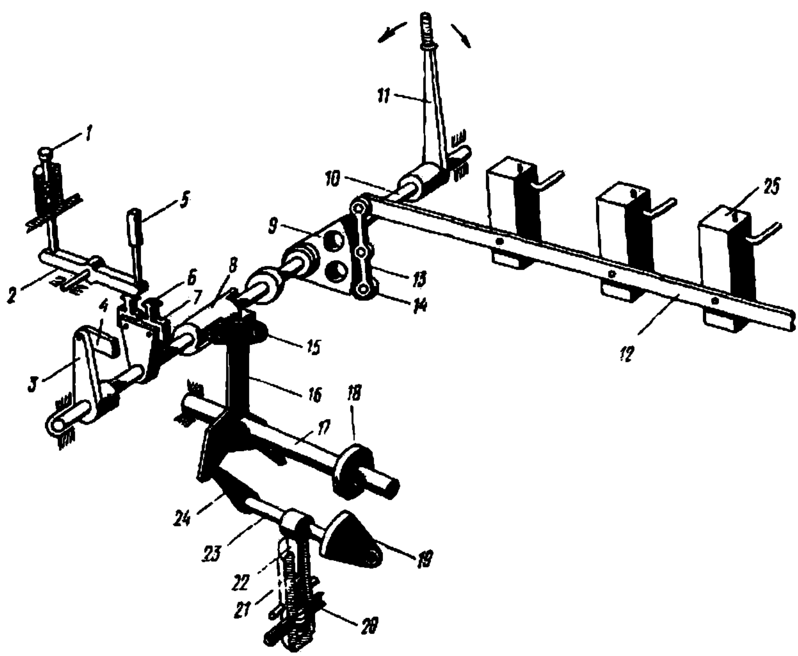


Fig. 100. Starting and reversing gear of ДР 30/50 diesel engine

1—emergency shutdown pushbutton; 2, 3, 13—levers; 4—tie rod; 5—pilot valve; 6—tappet; 7—bellcrank; 8—cylindrical interlocking cam; 9—fuel quadrant; 10, 17, 23—shafts; 11—control handle; 12—fuel control rack; 14—roller; 15—pin; 16—forked lever; 18—cam; 19—quadrant; 20—air pipeline; 21—interlocking valve body; 22—stem; 24—detent; 25—fuel-injection pump

Consider the starting and reversing gear of a ДР 30/50 marine diesel engine employing a turnable distribution block and symmetrical fuel cams. Fitted to shaft *10* (Fig. 100) there are regulating handle *11*, fuel quadrant *9*, cylindrical interlocking cam *8* with two grooves, bell-crank *7* and lever *3*. Fuel quadrant *9* is arranged to shift the control rack *12* of fuel-injecton pumps *25*, by way of lever *13* with rooler *14* at an end, into a position corresponding to the

setting of regulating handle *11* when this has been shifted from the "STOP" position into the "RUNNING AHEAD" or "RUNNING ASTERN" position.

The cylindrical distribution block of the starting air distribution valve is turned from one control position into the other with the aid of lever *3* and tie rod *4*. Two cams hinged to bell-crank *7* serve to open pilot valve *5* through the intermediary of horizontal lever *2* in starting the engine. The pilot valve can also be opened with the aid of emergency pushbutton *1*.

In starting the engine, regulating handle *11* is shifted from the "STOP" position by an amount causing the cylindrical distribution block to turn to the position appropriate to the desired direction of crankshaft rotation. Next, regulating handle *11* is shifted further so as to enable appropriate cam *6* to open pilot valve *5* with the aid of lever *2* and, as a result, to open the starting air master valve. This feeds air to the starting valves of the engine cylinders, starting air distribution valve and interlock. The air pressure applied to the pilot plungers of the starting valves causes these to open in accordance with the firing order, the opening sequence being controlled by the starting air distribution valve. For a failure-free starting, fuel quadrant *9* is to be set with the aid of regulating handle *11* so as to provide for a fuel delivery equalling 40% of the full-stroke one. As cylinders begin to fire, regulating handle *11* is shifted further into an operating position appropriate for carrying the engine load. After that the pilot valve is closed and the starting gear disengaged.

To reverse the direction of crankshaft rotation, the regulating handle is returned to "STOP" before being shifted into the position appropriate to the desired direction of rotation.

This causes lever *3* in conjunction with tie rod *4* to change the position of the distribution block in the starting air distribution valve. Further sequence of events is the same as indicated above. When the fuel delivery is being cut out, tappet *6* pivots, while maintaining contact with horizontal lever *2*, without opening pilot valve *5*. Regulating handle *11* must never be shifted into an operating position before the interlocking gear has performed its function while the engine is running in the direction corresponding to the position of the handle at "RUNNING AHEAD" or "RUNNING ASTERN".

The reversing gear of two-stroke diesel engines described above is simple and trouble-free in operation. The operation of opening the pilot valve is illustrated in Fig. 101.

Four-stroke diesel engines, timed in a different way for running ahead and astern, employ reversing gears arranged to shift axially the camshaft having two sets of cams which impart motion to the inlet and exhaust valves and the fuel-injection pumps. The axial

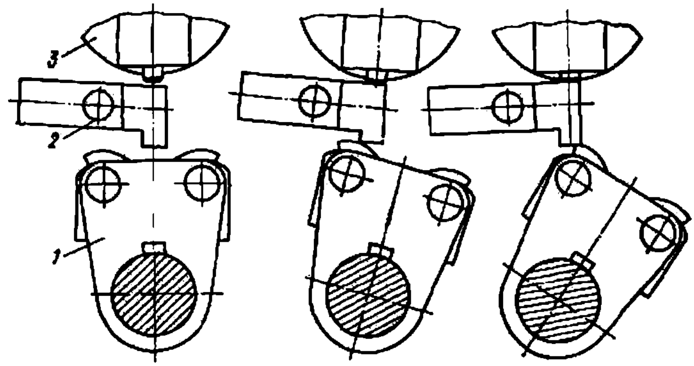


Fig. 101. Opening sequence of pilot valve on Д and Д 30/50 engines
1—lever with cams; 2—horizontal lever; 3—pilot valve

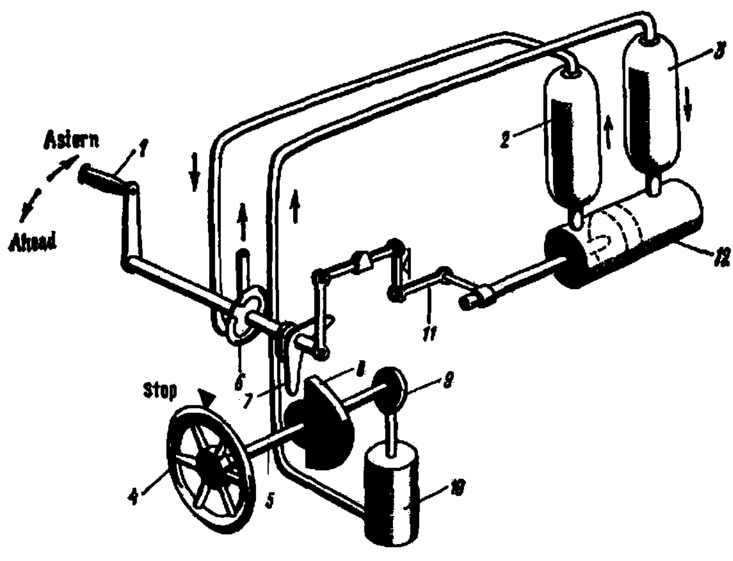


Fig. 102. Operating diagram of reversing gear
1—ahead and astern handle; 2, 3—reversing air cylinders; 4—starting handwheel; 5—axle; 6—interlocking gear; 7—bell-crank; 8—interlocking cam; 9—cam; 10—starting air master valve; 11—intermediate lever; 12—air motor

shifting of the camshaft is performed with the aid of a bell-crank.

A schematic diagram of a reversing gear is depicted in Fig. 102. To reverse the direction of crankshaft rotation, ahead and astern handle *1* is shifted to "AHEAD" or "ASTERN" positions, as required. Consequently, a starting air line becomes connected to reversing air cylinders *2* and *3*. When reversing handwheel *4* sharing the same shaft with cam *9* arranged to open the starting air master valve is turned to the "REVERSE" position, air is admitted into either of the reversing cylinders depending on the desired direction of rotation. From the reversing cylinder air enters an air motor which serves to shift the camshaft as required.

Further rotation of the handwheel admits air into the starting valves and sets the engine into operation.

The reversing gear with two sets of cams provides for any timing of the engine and for any timing of the fuel delivery in running astern. It may be used on diesel engines of all types. To maneuver the engine from running ahead to running astern, it is necessary to change the set of cams in operation. To that end, the rollers of the valve tappets are to be lifted clear of the cams, the camshaft must be shifted axially and the rollers lowered into contact with the other set of cams. A reversing shaft with eccentrics serving as fulcrum pins of the valve leverage is used to lift and lower the rollers.

80. Interlocking of the Starting and Reversing Gears

The function of the interlocking gear is to assure that the engine runs in the desired direction and no fuel is injected prior to the completion of the reversing operation. The engine can be started only after the interlocking gear has performed its function, i.e. the reversing gear has been set into the position for running ahead or astern.

An interlocking gear provides for correct functioning of the reversing gear and assures safe sailing of the ship. It is an indispensable element of the reversing gear of all direct-reversing diesel engines in marine applications. Most of the modern interlocking gears are of the mechanical type which is simple and reliable.

The interlocking gear of a ДР 30/50 diesel engine (Fig. 103) incorporates cylindrical interlocking cam *2* operating in conjunction with forked lever *4* which is running fit on shaft *5*—an extension of the camshaft. The forked lever carries rigidly-attached friction ring *1* at its lower end. Detent *6* and friction quadrant *7* are fitted to fulcrum pin *8* which, in its turn, is pivotally attached to the rod of interlocking spool valve *9*. When starting air is admitted into the spool valve through line *10*, causing the rod to lift, quadrant *7* comes abutting against friction ring *1* so that detent *6* turns forked lever *4*. As a result, pin *3* at the end of the forked lever disengages cylindrical interlocking cam *2*, enabling the regulating

handle to be set into a position providing for fuel delivery. If the reversing has been done in a correct way, the engine begins operation in the reverse direction. The camshaft also reverses the direction of rotation and so does friction ring *1*, causing quadrant *7* to turn integrally with detent *6* so as to induce the forked lever to turn as well. The upper pin of this lever disengages interlocking cam *2*, enabling the regulating handle to be shifted into a position appropriate to a given engine load.

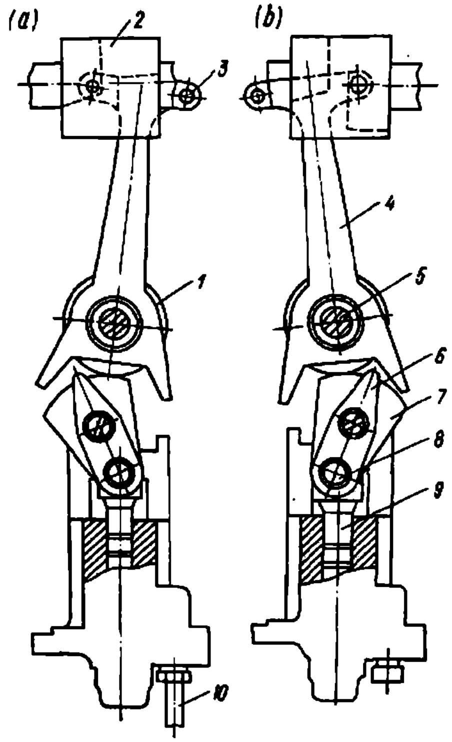


Fig. 103. Interlocking gear of ДР 30/50 diesel engine (*a*) fuel delivery interlocked; (*b*) interlock cleared and fuel delivery beginning

On somewhat different lines is arranged another mechanical interlocking gear used, for example, on the Sulzer 5SD-72 diesel engines producing 3 300 kW (4 500 hp) at 125 rpm. Referred to as a friction-type gear, it has detent *2* (Fig. 104) pressed against collar *4* of the tachometer shaft due to the

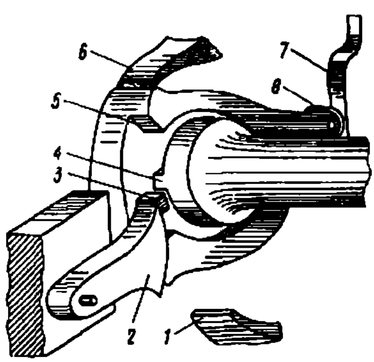


Fig. 104. Interlocking gear used on Sulzer diesel engines

pressure of the air admitted by way of a spool valve so that a projection of the detent engages a projection of collar *4*. When the crankshaft rotates in a direction opposite to the desired one, the projection of the collar presses the projection of the detent against internal flat *5* of an interlocking yoke. Missing the projection of collar *4* every half-revolution of the crankshaft and restoring then engagement, the projection of detent *3* slides inside the yoke while a roller is out of contact with cam *7*. When

the crankshaft is rotating in the desired direction, the projection of detent *3* is acted upon by the projection of collar *4* with the result that it clears the last-named projection and comes abutting against internal flat *5*. Responding to that action, the flat displaces towards stop *1* or *6* while roller *8* comes off cam *7* and causes the fuel-injection pumps to operate.

The engine controls may be interlocked with the engine order telegraph and barring gear. This assures that the engine controls are always in a position corresponding to the order from the bridge. Also no starting of the engine will be possible while the barring gear is engaged because of the interlocking gear cutting off the supply of air to the starting valves.

81. Starting Aids. Starting Gear Malfunctions

Trouble-free functioning of the starting and reversing gear is a factor influencing ship's maneuverability to a considerable extent. However, since the starting air expands on being admitted into the engine cylinders, its temperature lowers. This produces a cooling effect, inviting difficulties in starting. The situation may be aggravated by significant wear on the cylinder liners and piston rings, leaky inlet and exhaust valves, poor performance of the fuel-feeding equipment. An expedient to be resorted to, in order to minimize the cooling effect of the expanding starting air, is to keep the cylinders, pistons and fuel equipment in good repair. Apart from that, a good many of marine diesel engines are provided with means of heating the engine or its combustion chambers directly before starting. So, on some small engines electric heating plugs may be used to preheat the incoming air when the ambient temperature is 278 K (5°C) or less. However, a high power consumption limits the use of heating plugs. The preheating of either the lubricating oil in special tanks or the engine as a whole by circulating hot water through the cooling system is a reliable method of rising the compression temperature in the engine cylinders so as to render the engine limber for starting. High-output marine diesel engines are commonly warmed up by the jacket water outflowing from the operating diesel-generator sets. Engines with pre-chambers having a large area of heat transfer make use of electric heater plugs which are connected across a storage battery at the instant of starting to produce an increase in the temperature inside the combustion chamber.

Ship's maneuverability depends on the performance of a system consisting of the main engine, screw propeller and ship's hull.

The period elapsed in bringing at rest the crankshaft directly coupled to a constant-pitch propeller after the fuel-injection pumps have been cut out varies with the ship's speed at the beginning of the maneuver. The reversibility of a ship is influenced by the retar-

ding torque of the engine which, in its turn, is decided mainly by the construction of the starting and reversing gear. The period of retardation is dependent upon the initial ship's speed, propeller characteristics and some other factors.

The main propulsion plant of a ship equipped with a controllable-reversible-pitch propeller needs neither be stopped nor reversed in order to change the direction of ship's advance. The smooth and vibrationless reversing is completed in this case within a period lasting between 50 and 85% of the time required to carry out the engine reversing maneuver. Controllable-reversible-pitch propellers improve ship's maneuverability and the lifetime of the propulsion plant because less starting operations are required in this case.

A defect coming about in the course of a starting or reversing operation may incur a breakdown. Not infrequent are cases where the crankshaft responds to the opening of the starting air master valve by making barely one incomplete revolution instead of going on to spin. The most likely cause of the defect is either low starting air pressure or blocked air lines connecting the starting valves to the distribution valve. The remedy is to check pressure and recharge the starting air cylinders if the pressure is low or to blow off the lines with compressed air. A leaky starting valve which may stick or become defective in some other way is betrayed by the rising temperature of the pipeline connected thereto. The malfunctioning valve is to be taken apart, ground and checked for freedom of movement. Most susceptible to damage is the starting air master valve.

The reversibility and easy starting of turbosupercharged engines are dependent, among other things, on the performance of the turbine used to drive the blower. Correct timing of the starting valves which is determined experimentally by the builder is also a factor assuring easy starting on a minimum amount of air used.

REVIEW QUESTIONS

1. What are the stages of starting a diesel engine before it is running at the normal operation load?
2. What are the main components of the starting air system used on a direct-reversing diesel engine?
3. What purpose does the starting air master valve serve?
4. What is the function of a starting valve?
5. What is the means of opening the starting valves in accordance with the firing order of the engine cylinders?
6. What types of starting air distribution valves are used on marine diesel engines?
7. How is a diesel engine started electrically?
8. What is the difference between the reversing procedure of a two-stroke diesel engine and that of a four-stroke one?

9. What purpose does the interlocking gear serve?
10. What steps are taken to speed up the starting of a diesel engine?
11. What conditions assure the failure-free starting of a diesel engine?

Chapter XIV

CLUTCHES, REVERSING GEARS, COUPLINGS

82. Reversing Clutches and Hydraulic Means of Their Control

To disconnect the shafting from the propulsion plant for reversing the direction of travel or during a period of idling, ships are commonly equipped with clutches and couplings. Marine practice knows clutches (couplings) of various kind such as, for example, tooth-type clutches,

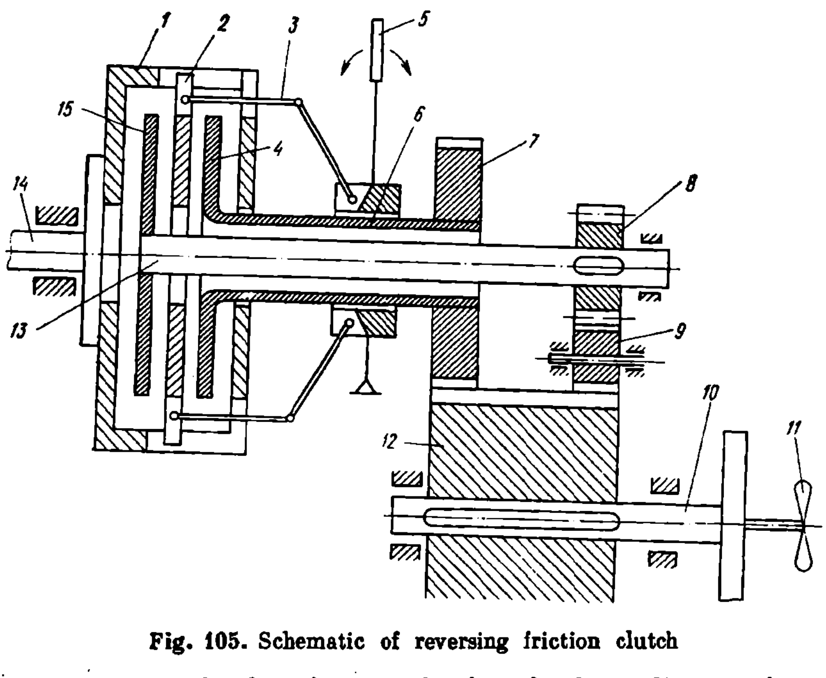


Fig. 105. Schematic of reversing friction clutch

pneumatic tire clutches, friction clutches, fluid couplings, and electromagnetic clutches. Each type is designed to meet the requirements of a particular type of ship and propulsion plant. So, pneumatic tire clutches find application on installations where one has to cope with misalignment of the shafting; a fluid coupling is expe-

dient in twin installations when two or more engines are used to drive the propeller through a common reduction gear unit. Some of the disengaging clutches serve also as reversing mechanisms.

Consider by way of example a reversing friction clutch found on board vessels powered by a low-output unirotational engine. It consists of clutch *1* proper (Fig. 105) made up of two discs *4* and *15* arranged to transmit power in running ahead and astern, respectively. Ahead disc *4* is fitted to hollow shaft *6* passing through which is shaft *13* carrying astern disc *15*. Gears *7* and *8* fitted to the ends of

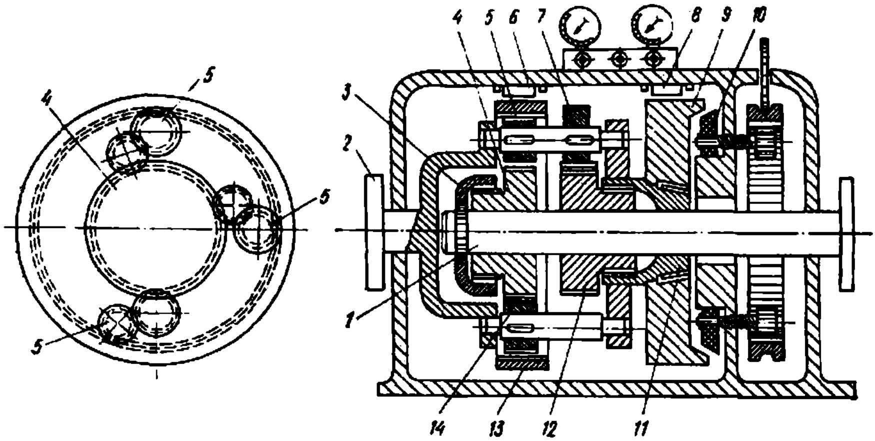


Fig. 106. Remotely-controlled clutch doubling as reduction and reverse gear, type NVD-24 diesel engine

shafts *6* and *13*, respectively, form a reduction gear unit transmitting the power from crankshaft *14* to the propeller shaft. Pressure plate *2* interposed between discs *4* and *15* can be shifted axially with the aid of leverage *3* actuated by control lever *5* and collar *6* sliding along hollow shaft *6*. An axial displacement of pressure plate *2* causes it to become pressed against either ahead disc *4* or astern disc *15*. In operation, when the control lever is shifted to neutral, discs *4* and *15* are rotating idle. A setting of the control lever into the position for running ahead, causes pressure plate *2* to contact ahead disc *4* of hollow shaft *13* so that power is transmitted to the propeller shaft through gears *7* and *12*. Should control lever *5* be shifted into the position for running astern, pressure plate *2* contacts disc *15* with the result that power is applied to propeller shaft *10* by way of reduction gears *8* and *12* as well as idler gear *9* which serves to reverse the direction of propeller rotation *11*. The clutch described is used on the 6ЧСП 15/18 and 6ЧСП 12/14 diesel engines as the reduction and reverse gear unit.

Some of the clutches used as reduction and reverse gearing are fitted with hydraulically-operated means of remote control as this is exemplified by the NVD-24 diesel engine (Fig. 106). Gear cluster *4* fitted to reducer input shaft *1* engages movable tooth-type clutch *3* along with three pairs of planet gears *5* and *7* which serve to transmit the power from the crankshaft to the propeller shaft in co-operation with three idler gears *14*. While each of planet gears *5* engages gear ring *13*, via driven shaft *1*, the planet gears *7* are in mesh with sun gear *12* which is also in contact with clutch *11* and brake pulley *9*. For running ahead, hydraulically-operated brake blocks *8* applied to pulley *9* arrest sun gear *12*, enabling planet gears *7* to orbit it. At the same time, gear cluster *4* transmits rotary motion to planet gears *5* which set into motion, in conjunction with idler gear *14*, gear ring *13* by way of their spindles. The result is that output shaft *2* linked to the propeller shaft is set rotating in the direction opposite to the rotation direction of input shaft *1* at a speed ratio of 1:2. Spool valves are used to control the motion of brake blocks *6* and *8* which are also provided with springs to keep them released during idling. Brake pulley *9* may be arrested with the aid of hand-operated device *10*.

83. Fluid Couplings

Fluid couplings have gained wide-spread recognition on marine diesel engines by virtue of their assets which are as follows:

—freedom from rubbing parts and, as a result, freedom from wear;

—operation of the output shaft independently of the rotation of the input shaft;

—continuous operation of the engine even in the event of a transient fouling up of the propeller so that the engine and shafting are safeguarded against breakdowns;

—high efficiency (96-98%);

—noiseless transmission of the torque;

—gentle starting and smooth acceleration of the ship;

—freedom from hazardous torsional vibration of the shafting, for the fluid coupling subdivides it into two self-contained systems.

For operation, a fluid coupling relies on a continuous interaction of an input member, or impeller, and an output member, or runner, which form a single unit.

Referring to Fig. 107, the impeller set into motion by the engine is shown at *5* and the runner at *6*. The impeller transforms the mechanical energy of the engine into the kinetic energy of a circulating fluid, producing a certain head. The runner transforms the kinetic energy of the fluid into the kinetic energy of the propeller shaft, or reduction gear *10*, imparting rotary motion thereto. In other words, the runner uses the head produced by the impeller.

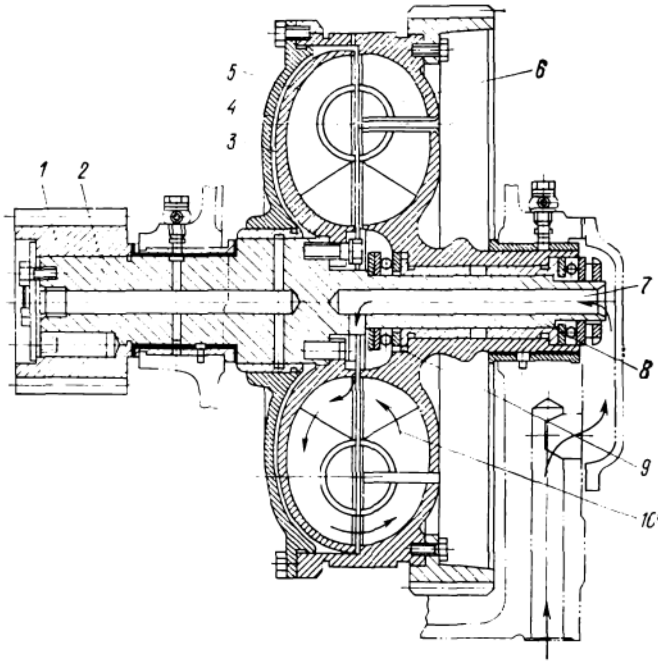


Fig. 107. Fluid coupling

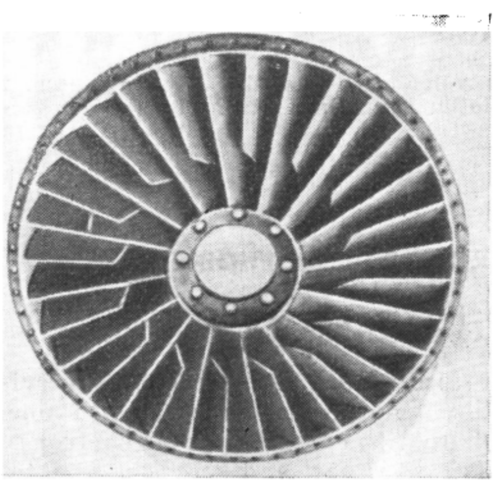


Fig. 108. Impeller of fluid coupling

When toroidal casing *5* of the input shaft is set rotating, the fluid entering the coupling through opening *7* begins to circulate over a closed circuit from the impeller to the runner and back to the impeller, causing toroidal casing *6* of the output shaft also to rotate owing to internal frictional forces. Thus, the energy is being transmitted from the input shaft to the output shaft by way of the fluid. Unavoidable losses of the energy due to the friction between the particles of the fluid cause a slip of the output shaft with respect to the input one amounting to 2 or 3%. This reduces the efficiency of the fluid coupling by 3%.

The toroidal space of impeller *5* is divided into compartments by a series of flat radial vanes *3* and is enclosed by cover *4* (Fig. 108). Toroidal casings *5* and *6* of the input and output shafts *2* and *8*, respectively, are fitted with clearance of 5-12 mm between them. Journal ball bearing *9* is supplemented by thrust ball bearings to prevent axial displacements of the casings.

The fluid coupling assures reliable connection and enhances the maneuverability of the propulsion installation, catering for the reversing of the propeller from the control console by cutting in and out the engines which rotate in different directions and drive a common propeller shaft through gear *1*. It also protects the diesel engine against impact loads arising when the ship is traversing ice.

REVIEW QUESTIONS

1. What purpose does a clutch serve?
2. Discuss the operation of a reversing friction clutch.
3. What purpose do fluid couplings serve? Discuss their assets and limitations.

Chapter XV

REMOTE CONTROL OF DIESEL ENGINES

84. Mechanical Remote Control Systems

Modern marine propulsion plants can be supervised in operation not only from the engine room; they are also amenable to remote control exercised from the wheelhouse or control room. The equipment used in this case is part and parcel of an integrated system of propulsion plant automation. Assuring operation without relaying orders, it reduces the crew complement, improves labour conditions and enhances ship's maneuverability.

A mechanical remote control system is the most simple one. It comprises shafts fitted with gears, leverage and cables which are all arranged so as to actuate the engine controls from the wheel-

house. Gears and levers operate with a minimum amount of lost motion resulting mainly due to wear on pivots. Cables display a larger lost motion resulting from their residual deformation set up after a lengthy period in service. To keep the lost motion within an allowable limit, the maximum length over which a mechanical remote control system can operate is 20 m, which is its obvious limitation.

A schematic arrangement of the remote control system used on a 6NVD-48 diesel engine is depicted in Fig. 109. Sliding sleeve *5* with cogs at either end face is arranged to displace axially, using control lever *4*, depending on the desired direction of crankshaft rotation. When set into either of its extreme positions, the sleeve has its cogs in mesh with appropriate notches in sprockets *6* and *12* fitted to shaft *7*, sliding along which is sleeve *5*. Roller chains *13* passing over sprockets *6* and *12* are connected to cables *14* which actuate pulleys *20* and *18* on the engine room console. Guide slots *1*, *2*, *3* (see the inset) provided on the wheelhouse console permit the shifting of control lever *4* into appropriate positions so that pin *10* at its end displaces along annular grooves *8*, *11* and longitudinal groove *9*. The movements of control lever *4* are repeated by ahead and astern handle *19* and regulating handle *17* in the engine room. Control cables *14* pass over rollers *15* and are adjusted for tension with the aid of turnbuckles *16*.

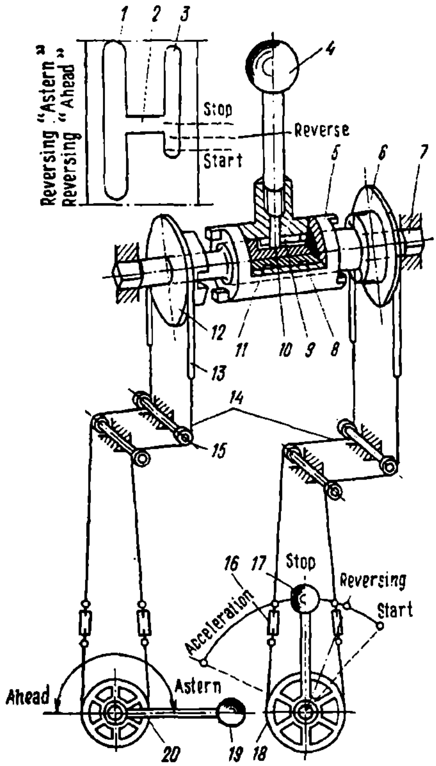


Fig. 109. Schematic of reversing gear in 6NVD-48 diesel engine

85. Pneumatic, Electromechanical and Pneumatomechanical Remote Control Systems

A pneumatic remote control system uses compressed air to actuate the engine controls, control air pressure being applied in response to shifting a control lever on the console located either in the wheel-

house or on a bridge wing of a motor ship. Simple and reliable as it is, the pneumatic remote control system can be applied to engines of various types. A reversing maneuver initiated from the wheelhouse commonly lasts between 4.5 and 7 s, provided the initial starting air pressure is 118×10^4-236×10^4 N/m^2 (12-24 kgf/cm^2). However, its performance is efficient at a maximum distance of 30 m, for the surges of air may delay the progress of impulse air. Also high impact loads are likely to be set up in the air pipelines when the system is set into operation. Apart from that, the condensate gradually accumulating in the pipelines may interfere with the operation of the equipment to a point when a starting or reversing will worsen or become impossible.

Pneumatic remote control is a feature of the 4ДР 30/50 diesel engine producing 294 kW (400 bhp) at 300 rpm. In addition to an engine room desk used in emergency, a control lever on the wheelhouse console permits the starting, shutting down, reversing and speed control of the diesel engine.

An electromechanical remote control system consists of a mechanical linkage and an electrical switch, the former serving to control diesel operation from the wheelhouse, and the latter, to cut in and out an electrical motor arranged to actuate the fuel rack. Such a system is employed on the tugs of the "Khariton Laptev"-class powered by two S-275L diesel engines with an output of 294 kW (400 bhp) at 550 rpm. The time lag of the remote control link is 2-3 s, and the period of a reversing maneuver is 14-17 s. However, the 20-metre mechanical linkage appears to be bulky and therefore difficult to install on board the ship, let alone that it gives rise to the time lag.

Most of the contemporary engine remote control systems employ both mechanical linkage and pneumatic equipment. The orders "START", "STOP" and "FUEL DELIVERY" are initiated with the aid of a control lever connected to the mechanical linkage in the form of chain-and-sprocket drives and cables. For reversing, the recourse is made to the combined mechanical and pneumatic linkage—with the control lever being shifted to "REVERSING". Speed regulation is effected in the same way by shifting the fuel rack or exerting an action on the governor. The control console is located either in the wheelhouse or, if towing operations are in prospect, on the upper bridge deck as this is the case with the "Poltava"-class tugs equipped with two Cupper-Bessemer diesel engines, each developing 478 kW (650 bhp) at 400 rpm.

The advent of highly maneuvreable motor ships depends on further improvement of the starting and reverse gear designed for use on marine diesel engines.

86. Remote Control of Propulsion Plants With Controllable Reversible-Pitch Propellers

Marine propulsion installations with controllable reversible-pitch propellers have recently gained wide-spread recognition, particularly on ships of the fishing fleet. So, for example, the freezer trawlers of the "Mayakovsky"-class are all equipped with controllable-pitch propellers and so are many trawlers of the "Okean", "Tropik" and other classes. Controllable-pitch propellers assure efficient operation of the engine sustaining fractional loading as this is the case with trawling operations. For good engine economy, the propeller

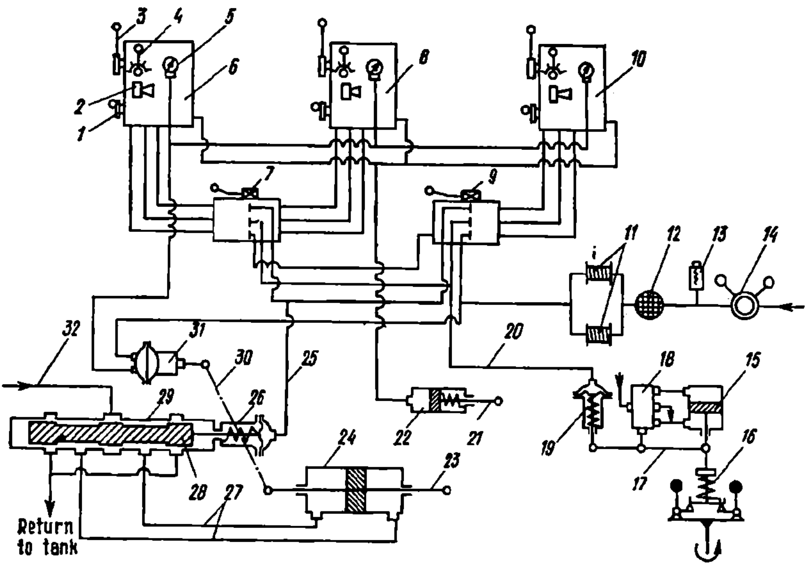


Fig. 110. Circuit diagram of remote-control system used on diesel engine coupled to controllable-reversible-pitch screw propeller

pitch should be an optimum match to the engine speed. This aim is better attained through a programmed system of automatic control.

Depicted in Fig. 110 is the schematic diagram of a remote control arrangement which employs pneumatic and hydraulic equipment to change the propeller pitch simultaneously with the engine speed in accordance with the programme. It finds application in trawlers of the "Mayak"-class powered by the 8NVD-48 main engine.

As is seen, there are three remote control consoles *6*, *8* and *10* located on the bridge, in the wheelhouse and engine room, respectively, each being fitted with a single control handle. Control air is

taken from the starting air receivers by way of a pressure-reducing valve reducing the pressure from 280×10^4 N/m^2 (30 kgf/cm^2) to 37×10^4 N/m^2 (4 kgf/cm^2), passing then through filter *12* and moisture trap *11*. Safety valve *13* is interposed between the filter and moisture trap. Each control console is provided with pitch and speed set-point devices of the pneumatic type linked to control handle *3*. A preset programme may be changed with the aid of pitch-adjusting handle *4* or speed-adjusting handwheel *1*. Hooters *2* provided at each console initiate alarms in case of an engine overload. To change over from one control console to another, use is made of air-operated interlocking valves *7* and *9* located in the wheelhouse and engine room, respectively. The impulse air admitted into line *25* by turning the control handle *3* on the wheelhouse console causes actuator *26* with a spring-loaded membrane to shift spool valve *28* of servomotor *29*. The spool valve directs oil from pressure line *32* into hydraulic motor *24* via one of lines *27* depending on the desired direction of displacement of its piston linked to the pitch-controlling mechanism of the propeller by way of rod *23*. At the same time, the feedback lever acts upon the actuator so that it sets the spool valve of the servomotor into its midmost position with the result that the propeller blades are locked in the appropriate position. Furthermore, the feedback lever also acts on set-point device *31* which changes the air pressure in the line proportionally to the blade angle, pressure gauges *5* serving to monitor the pressure.

The speed control is effected with the aid of pressure transducer *19*, spool control valve *18* and air motor *15*. A change in the impulse air pressure in line *20* causes a displacement of the air motor piston arranged to act on spring *16* of a variable-speed governor. Lever *17* provides the feedback link.

The engine protection system consists of air motor *22* connected to the fuel control rack through its spring-loaded rod *21*. In emergency, compressed air admitted into air motor *22* by turning handwheel *1* causes the air motor to shift the fuel rack so as to cut out fuel delivery.

REVIEW QUESTIONS

1. What purpose does an engine remote control system serve?
2. What remote control systems are used in marine practice?
3. What are the assets and limitations of the existing remote control systems?

Chapter XVI

AUTOMATION OF MARINE DIESEL ENGINES

87. Automatic Speed Regulation

Engine power varies directly with the amount of the fuel injected into the engine cylinders. The lessening of load coming on the engine shaft calls for the reduction in the quantity of the injected fuel, or otherwise the engine speed will increase. Should the engine load exceed the output, the rate of crankshaft rotation will decrease to a point when the engine will stall. This implies that the torque the engine develops under steady-state conditions must be equal to work done by the resistance forces.

The amount of fuel injected per cycle is varied with the engine load at a given engine speed by means of governors. A governor of a main engine directly coupled to the propeller is set to maintain a given engine speed irrespectively of the engine load. To that end, the governor intended for installation on such an engine must be capable of maintaining any speed set from the control console within the range of low and full speed. Such a governor is referred to as a variable-speed unit.

Auxiliary engines are bound to sustain variable loadings without any change in their speed, and must be fitted with governors capable of catering for the functioning of one or several generators connected in parallel. This requirement is of particular importance for diesel-alternator sets. Thus, a governor suitable for this application must maintain a constant engine speed over the range between idling and full load, irrespective of load fluctuations. It is termed a constant-speed governor. Some of the high-speed marine diesel engines of the automotive type linked with the propeller through a reverse clutch are provided with governors capable of maintaining two speeds: idling and maximum. They are known as two-speed governors. In recent times these governors are replaced by the variable-speed type reputed for its versatility.

Depending on the design, governors fall into two categories referred to as self-actuated and pilot-actuated devices.

In a self-actuated governor, the sensing element is movably linked to the final control element (fuel-injection pump) through an actuator (fuel rack), supplying by itself the energy required to shift the actuator. Finding application on low- and medium-power output engines, this type consists of the governor shaft, crankshaft-driven governor weights of various configuration, the governor sleeve, springs and leverage. The weights of the sensing element are commonly made in the form of two L-shaped masses, and the governor springs are

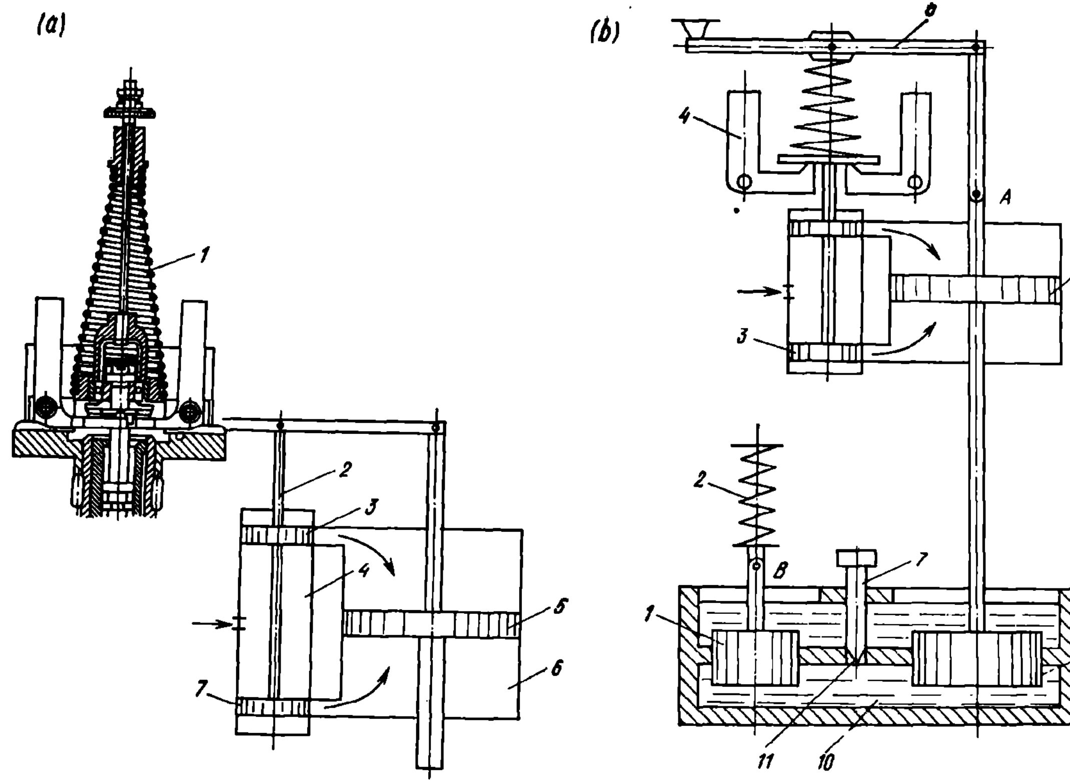


Fig. 111. Speed governor

arranged so as to set at balance the centrifugal force on the weights. A requisite engine speed is set in operation by changing the spring tension. The governor obtains drive from the camshaft through a gear attached to the shaft carrying the weights. The control action of the governor sleeve is transmitted to the fuel-injection pump through levers and tie-rods. The disadvantages of self-actuated governors are their size, growing directly with the engine output, and difficulties experienced in obtaining stable performance.

In a pilot-actuated governor, the servomotor doubling as an amplifier is used to overcome the resistance of the linkage and amplify the control signals produced by the sensing element. It also incorporates an antihunting device slowing down the displacement of the actuator in good time. In use are hydraulic, pneumatic and electric servomotors.

Pilot-actuated governors, in their turn, are classified as unity feedback and vanishing feedback, also referred to as proportional-plus-integral governors.

Consider details of the pilot-actuated governor design. Referring to Fig. 111*a*, the hydraulic servomotor consists of cylinder *6* with reciprocating power piston *5* acting on a tie-rod connected to the fuel-injection pump rack. Spool valve *7* linked to sensing element *1* by means of levers serves to admit oil into, and discharge it from, the cylinder.

To assure steady regulation, pilot-actuated governors commonly employ a feedback device. Its necessity has been proved by practical experience. Suppose, the load on an engine fitted with a pilot-actuated governor without feedback decreases suddenly, the crankshaft sets about sharply to gain speed with the result that the governor weights move apart, uncovering an upper port in sleeve *4* of the spool valve through the intermediary of rod *2*. This action admits oil into servomotor *6* which causes its piston *5* to move so as to reduce fuel delivery, i.e. downwards in the case under consideration. However, the engine speed will go on increasing, causing the governor weights to move farther apart, because of the fuel delivery being somewhat higher than it is required in order to carry reduced loads. Further downward travel of piston *5* sets the requisite fuel delivery, yet the piston continues its displacement due to inertia, carrying on with its delivery-reducing assignment. The engine responds by slowing down, and the governor weights impart motion to the spool valve until it interrupts the flow of oil into servomotor *6* and piston *5* is brought at standstill. At this instant, the fuel delivery will again fail to match the given load, rendering a repetition of the regulation process imminent. In other words, the governor weights will start approaching each other, causing piston *5* of the servomotor to displace in the direction appropriate for an increased fuel delivery.

Thus, in the absence of a provision for a feedback, regulation becomes a nonattenuating process, and the rate of crankshaft rotation always a figure differing from the set point. This implies that the governor is to be supplemented with a means of arresting the spool valve in good time during its travel upwards or downwards so as to reduce engine speed fluctuations. This is the feedback bringing the piston of the servomotor at a halt as soon as it has displaced through a distance corresponding to the change in engine speed. A unity feedback comprises a leverage returning the spool valve into its midmost position time and again. However, a vanishing feedback displays by far a better performance in this respect. It is carried out with the aid of what is called a proportional-plus-integral element (PI-element) existing in a number of types. If a PI-element is used, regulation begins on the same lines as this is the case with a unity feedback but then the PI-element cancels the feedback, enabling the engine to resume its original speed.

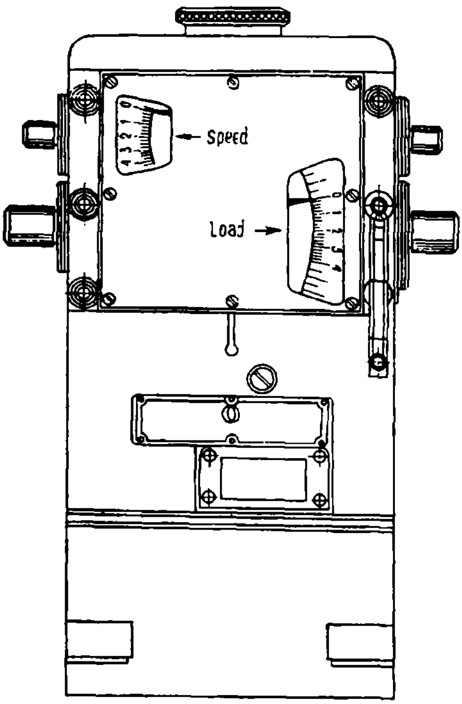


Fig. 112. Pilot-actuated variable speed governor

Consider by way of example the operation of a PI-element depicted in Fig. 111*b*. Shown at *8*, this element comprises a hydraulic cylinder with piston *9* and spool valve *1*. At an early stage of regulation, a control signal in the form of a displacement initiated by sensing element *4* is transmitted via lever *5* from point *A* to point *B* through the agency of the oil the PI-element is filled with. At a later stage of regulation, piston *6* displaces until spool valve *3* returns into its initial position and so does point *B* acted upon by spring *2* while oil *10* slowly flows from one space of the servomotor into the other by way of opening *11* partly covered by screw *7*.

A PI-element appears to be of particular value in those cases when a number of diesel-generator sets are operating in parallel and the speed of each generator is to be maintained in conformance with the

current frequency. For speed regulation use is also made of precision variable-speed governors of the centrifugal type which are pilot-actuated and employ a proportional-plus-integral feedback. Such a self-contained governor with a hydraulic servomotor and closed-circuit hydraulic system is depicted in Fig. 112; it is engine-driven through a splined shaft.

The advantages of pilot-actuated governors are rapid response, the application of full force to the actuator at an early stage of servomotor operation and compactness. On the other hand, this type is more complex as compared with self-actuating governors.

88. Automation and Integrated Mechanization of Marine Engines

Reliable operation of the systems controlling and supervising shipboard propulsion plants, which are capable of performing numerous functions accurately and at a high speed, is essential for the trouble-free running of a ship. The monitoring of propulsion engine performance, speedy trouble-shooting and elimination of defects call for manning the engine room with highly skilled personnel. However, a good part of their watch is spent on checking the instrument readings, coping with the gauging and sounding, and making entries into the engine room log, unless the propulsion plant is an automated one.

To eliminate the human factor and render the main engine more reliable and, consequently, efficient, there is a wide-spread recourse to shipboard automation and integrated mechanization. Their use is pivotal to decreasing the crew complement and introducing partially unattended engine-room operation. The degree of automation is indicated in each particular case by the period of unattended operation.

One of the basic modes of mechanized operation is remote control of the main engine and auxiliaries involving a completely automated starting up and reversing along with the supervision of the main engine parameters. Engine room automation of a varying degree is a feature of all modern ships, automation also gains ground in other spheres of ship handling.

An automatic control and supervision system is employed, for example, on the 22 690-tdw roll on/roll off ship "Komsomolsk" built by the Valmet Oy shipyard of Finland for the USSR. The two medium-speed engines with a total power output of 19 850 kW (27 000 hp) can be operated without watchkeepers in the engine room of the ship which has a length of 206 m and a breadth of 31 m.

The research vessel "Professor Bogorov" is equipped with an alarm system enabling the supervision of all engine room machinery from the central control room. The system has been designed on the principle of a continuous alarm scanner running through all points to

seek values which are outside their predetermined limits. On the main diesel, it monitors fourteen values vital for trouble-free operation. In the event of low oil pressure or high bearing temperature, visible and audible alarms are being initiated, urging the personnel to take appropriate steps in order to rectify the defect. An event log is automatically printed in red during the emergency, indicating the source and time of the defect.

Also in use are computerized systems of alarm scanning and monitoring. So, the 32 000-tdw petrochemical tanker "Maikop" powered by a Sulzer 6RND-76 engine developing 8 850 kW (12 000 hp) at 122 rpm has such a system, capable of monitoring 200 points, installed in her control room. An event log is also printed out in emergency when alarms are being given, indicating the date, time and source of the trouble.

Control consoles are provided with mimic diagrams helpful in speeding up the trouble-shooting and elimination of defects. Fitted with indicating lamps, these diagrams show to the engineer of the watch the state of the machinery in operation and enable the remote control of oil, fuel and water pumps. Other mimic diagrams are arranged to control the operation of valves.

Automation appears to be an expedient that assures the running of the engine in an optimum way and relieves the personnel of the necessity to keep the log. It also permits partially unattended engine-room operation. However, automation calls for manning the ship with highly skilled personnel, for the watchkeeper must be capable of trouble-shooting the automatic system in the event of failure and recondition it by plugging-in a new unit instead of the defective one.

Taking into account that there is a tendency to replace human beings by adequately reliable automatic equipment which fosters further progress in shipbuilding and improves labour conditions, the degree of automation to be adopted on board ships of each particular type is decided by the economic gain.

REVIEW QUESTIONS

1. How is the power output of a diesel engine made to match the load on it?
2. What speed governors are used in shipboard practice?
3. What is the function of the feedback in a speed governor?
4. Discuss the advantages of vanishing feedback.

Chapter XVII

INSTRUMENTATION AND ENGINE PROTECTION EQUIPMENT

89. Instrumentation

Engine operation is constantly monitored with the aid of instruments arranged mainly on the control console. Some of the instruments, such as tachometers, totalizing revolution counters, pressure gauges and thermometers, are used continuously. There are also instruments in transitory use for checking engine performance from time to time. Coming under this category are engine indicators, piston-type pressure gauges and portable tachometers. Often electrical instruments, gas analyzers, noise and vibration meters are used to ascertain the wear on components, determine noise levels, study the operating cycle and cope with various aspects of research.

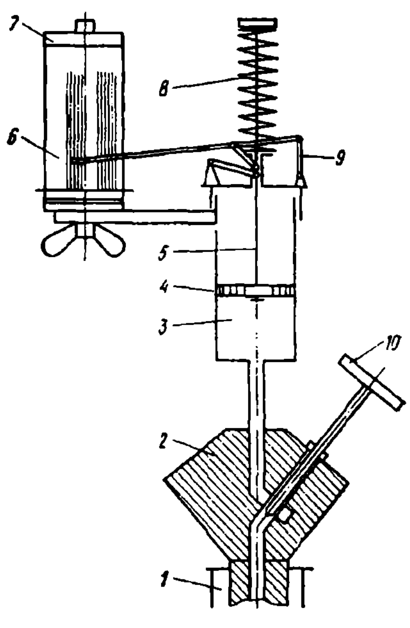


Fig. 113. Engine indicator

Consider the design features and operating principles of some instruments in transitory use.

Indicator diagrams showing the way the pressure in an engine cylinder is being changed depending on the piston motion are taken with the aid of indicators. From a diagram it is possible to ascertain the maximum combustion pressure, the indicated mean effective pressure and to check the valves for correct timing. Indicators for use on the engines operating at a maximum speed of 600 rpm employ cylindrical helical springs, those applied to high-speed engines developing a maximum combustion pressure of 686×10^4 N/m^2 (70 kgf/cm^2) feature a flat spring, for the inertia loads set up due to the high speed are likely to distort the indicator diagram if the spring is of the cylindrical type.

An engine indicator (Fig. 113) essentially consists of body *2* and driving gear *9* which, in its turn, comprises a cylinder *3* with piston *4*, pen arm *6*, spring *8* and chart drum *7*. The piston is linked with the pen arm through a leverage. The drum is revolved with the aid of a

cord attached at its other end to a crank or some other part of the engine whose motion coincides with that of the engine piston. Since the circumference of the drum equals only a fraction of the piston stroke, an appropriate reducing mechanism is used to enable the motion of the drum to coincide on a certain scale with that of the engine piston. Each indicator is supplied with assorted springs having various tensions which are used depending on the maximum combustion pressure in the engine cylinder at the instant of taking the diagram. Being fitted to indicator piston rod *5*, spring *8* functions as an extension type. The so-called scale, i.e. the pen stroke corresponding to a pressure of 9.8×10^4 N/m^2 (1 kgf/cm^2), is stamped at the bearing end of the spring along with the maximum pressure the spring is rated to sustain and the piston diameter in millimetres. The space *3* below indicator piston *4* is connected to the engine cylinder through a passage between 6 and 8 mm in diameter provided in body *2*. Three-way cock *10* permits the indicator to be connected either to the cylinder or with the atmosphere. Coupling nut *1* is used to fit the indicator to an indicator cock of the engine.

A flat-spring indicator is arranged on the same lines. The spring, made as a beam of uniform strength with a cross section varying so that the unit stress remains constant, is secured in place with the aid of a nut and cone. A ball provided at an end of the spring enables it to pivot about the indicator piston rod, operating thus in bending.

The indicator operates as follows. Once the space inside the engine cylinder is connected to the space below the indicator piston, the piston starts to reciprocate up and down integrally with the driving gear it is linked with. The combined rotary, i.e. horizontal, motion of the drum and the vertical movement of the pen will cause the latter to trace a closed indicator diagram. On measuring the area of the diagram thus taken and constructing a rectangle of the same area, we can find the so-called mean indicated pressure p_i which is given by the height of rectangle to the scale of the indicator spring.

What is called a draw card can be taken by disconnecting the indicator cord from the engine and pulling it by hand, obviously on connecting the indicator to all engine cylinders in succession. The pen will trace in this case a series of vertical lines the lengths of which will correspond to the compression pressures and maximum combustion pressures in the cylinders. The actual values of these pressures can be obtained by measuring the lengths of the lines with the aid of a scale rule.

A time-average pressure p_t is read off scale *9* of a piston-type pressure gauge which can be fitted to an indicator cock by means of coupling nut *1* (Fig. 114). When the indicator cock is set open, the gas pressure applied to piston *2* causes it to displace inside space *3* integrally with its rod linked with leverage *4* and spring *8*. The motion is transmitted to pointer *7* via gear *6* and toothed quadrant *5*

connected to one of the levers. A fulcrum pin fitted to which is the pointer has a flywheel at its opposite end which adds to the inertia of the leverage, eliminating thus pointer fluctuations. The piston-type pressure gauge is instrumental in tuning up an engine so that its cylinders equally share the load.

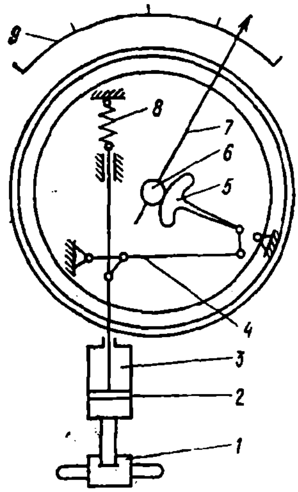


Fig. 114. Piston-type pressure gauge

The number of revolutions the crankshaft has made during a given period (e.g. on passage, between two successive overhauls or in order to ascertain fuel consumption) is registered by means of a totalizing revolution counter. Referring to Fig. 115, such a counter is made up of a plurality of gears equalling in number to the number of the orders used (units, tens, hundreds, thousands, etc.). The number of teeth on each gear is a multiple of ten. The rim of each gear bears digits from 0 to 9 and is fitted with pins, numbering twenty at one side and two at the other. Each pair of gears *1* and *2* engages gear *3* having four full-length teeth which alternate with four half-length ones.

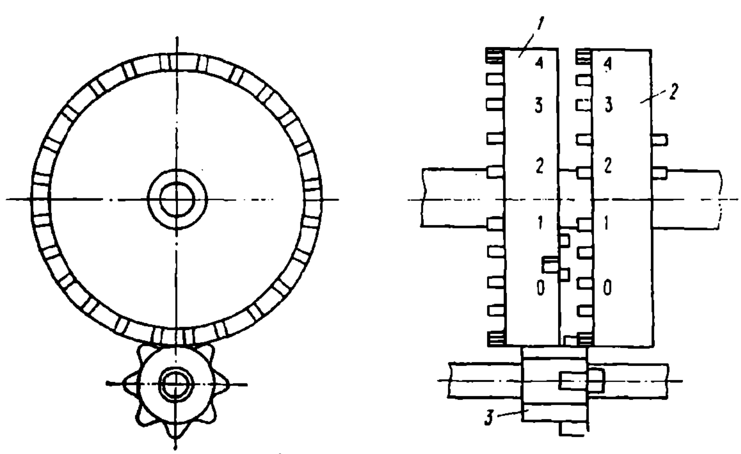


Fig. 115. Totalizing revolution counter

In operation, the gear denoting the units is rotating constantly while the gear serving to indicate the tens is held at standstill by

the full-length teeth of the meshing smaller gear. However, when the two pins engage the long teeth, the gear denoting the tens turns through an angle corresponding to two teeth, showing the digit which indicates the next ten revolutions.

The revolutions per minute of an engine are measured with the aid of an instrument termed tachometer. Depending on the design, distinction is made between centrifugal, electric and permanent-magnet-field tachometers.

A centrifugal tachometer (Fig. 116) relies for operation on a force exerted by weights when these move outward due to a centrifugal force coming thereon. Shaft *2*, rotating about which are the weights, is actuated by flexible shaft *1* or a chain-and-sprocket drive. The force corresponding to a certain position of the weights is balanced against helical spring *3* wound around shaft *2*, and the resulting deflection of pointer *7* across scale *5* calibrated in terms of the crankshaft speed is transmitted through tie rods and gear drives *4* and *6*. Friction in the movement may impair the accuracy of readings.

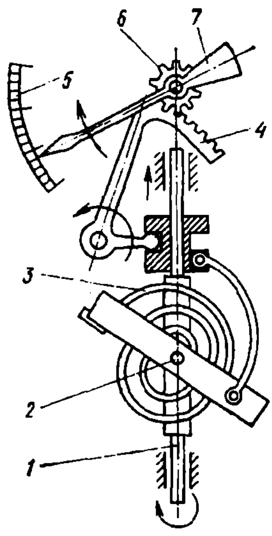


Fig. 116. Tachometer

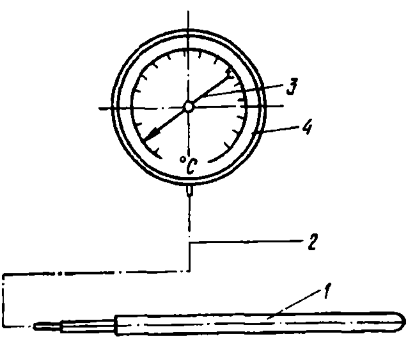


Fig. 117. Filled-system thermometer

An electric tachometer consists of a tacho-generator, a tacho-motor, a flexible shaft and connecting leads. The tacho-generator is a three-phase generator driven by the crankshaft through the flexible shaft, and the tacho-motor is a synchronous reaction motor set into motion by the current the tacho-generator produces in operation. When a drag cup made of copper cuts across the magnetic field set up by a magnet rotating integrally with the tacho-motor rotor, the current induced in the drag cup is sufficiently strong to cause a pointer to deflect. Electric tachometers assure accurate measure-

ments. They also enable the use of one tacho-generator in conjunction with a number of tacho-motors.

The operating principle of a permanent-magnet-field tachometer is based on an interaction of the magnetic flux produced by a rotary magnet with the eddy currents this flux induces in a conductor. A torque resulting from this interaction deflects the conductor integrally with a pointer through an angle varying directly with the rate of crankshaft rotation. This type employs a uniform scale, which is an asset. However, a susceptibility of permanent-magnet-field tachometers to temperature changes limits the sphere of their application on marine diesel engines.

The temperature of water, oil, fuel and air is measured with the aid of expansion thermometers, change-of-state thermometers and thermocouples. Expansion thermometers in the form of mercury-in-glass instruments are employed to monitor the temperature of jacket water and lubricating oil. Thermocouples are used to measure the temperature of exhaust gases. Their operation is based on a thermoelectric effect involving the setting up of a potential difference across a junction between two conductors in different metals with an increase in temperature. The potential difference is then measured with the aid of a sensitive instrument.

When there is a need in a remote indicating instrument, the recourse is to a filled-system thermometer coming under the change-of-state category (Fig. 117). Its operation is based on changes in the pressure of a liquid with temperature. The temperature-sensitive element consists of bulb *1* filled with a low-boiling liquid and placed in a medium whose temperature is indicated by pointer *3*. Any variation in the temperature of the medium changes the pressure in the bulb which is connected to pressure gauge *4*, calibrated in degrees Celsius, by way of capillary tube *2*.

90. Alarm and Engine Protection Systems

Automatic alarm monitoring and engine protection systems are designed to monitor the operation of an engine, warn the personnel, if any of the quantities monitored goes off-limits, and automatically shut down the engine in extreme cases so that consequential damage is reduced, if not nullified. So, for example, a warning is given in the event of high lubricating oil temperature or low jacket or outboard water pressure. However, a too low lubricating oil pressure or a drop in the jacket water pressure cause an automatic shut down of the engine.

The alarm monitoring system consists of sensing elements and alarm panels fitted with flashing lights and alarm horns or bells. Commonly monitored are the pressure and temperature in the lubricating and cooling systems, bearing temperature, engine load, fuel

line pressure, the temperature of the water cooling the pistons, the level of oil in the sump tank, etc.

In use are predominantly electric alarm monitoring systems. They are simple, dependable, impose no limitations on the distance between the units and permit the amplification and repeating of signals. Pneumatic equipment is used to a lesser extent.

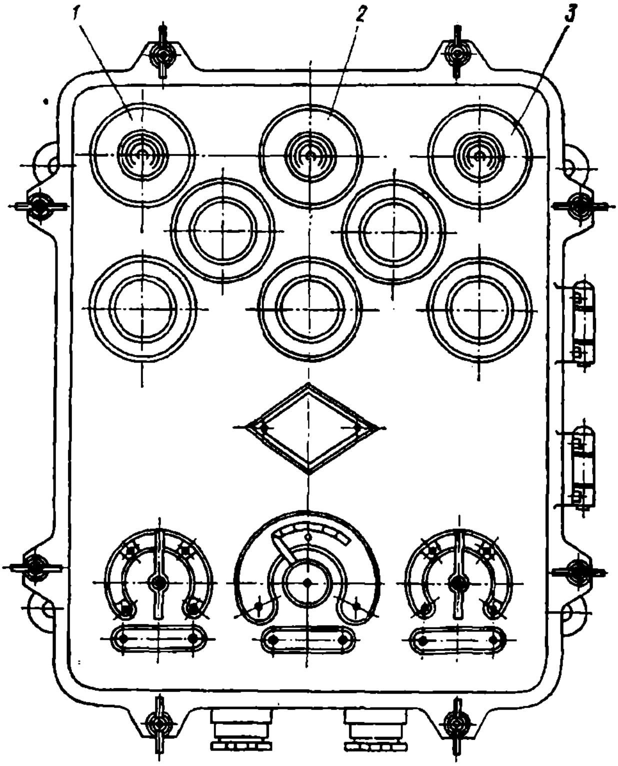


Fig. 118. Alarm panel

In case, for example, of a high jacket water temperature, an appropriate sensing element responds by closing the contacts of a final-control relay, energizing the circuits of red lamp *1* and indicating lamp *3* which are lit on the alarm panel (Fig. 118). At the same time, green lamp *2*, which is lit as long as the point monitored is inside the limit, becomes dark and an audible alarm is initiated. This can be then turned off. An inscription of the panel next the indicating light heralds the source of trouble.

In the event of a low lubricating oil pressure, the engine protection equipment takes over, shutting down the engine automatically and

preventing its starting on fuel before steps are taken to restore the pressure to normal. The equipment consists of a pressure-sensitive element arranged to stop fuel delivery by way of the fuel-injection pump actuator and of a latch for disconnecting the servomechanism serving to set the fuel-injection pump into the position of zero delivery.

Referring to Fig. 119, in operation, as long as the lubricating oil pressure is normal, pressure sensitive piston *4* remains in its leftmost position integrally with piston rod *5* so that springs *1* and *10*

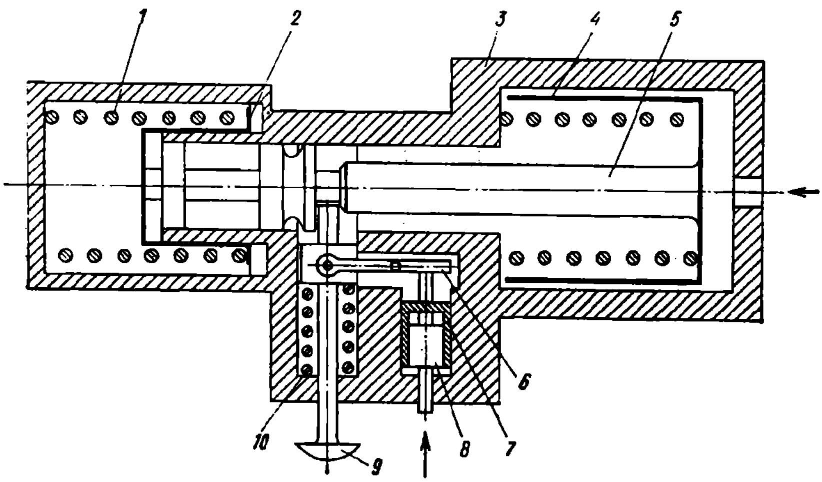


Fig. 119. Engine protection equipment

are, consequently, compressed. Stop *2* attached to casing *3* limits piston travel. Intermediate piston *7*, which is lifted integrally with a stop attached thereto by the air fed from the air receiver, keeps latch *9* in the retracted position, i.e. renders it disabled. Should the oil pressure decrease below the permissible level, the compressed springs displace piston *4* rightwards and move the latch downwards with the result that a lever located coaxially with the latch sets the fuel-injection pump rack into the position of zero delivery. When the engine has, consequently, come at rest, no air is fed from the receiver and the latch snaps into the piston rod due to the action of the spring. The resulting fixing of the latch facilitates the tuning up of the fuel-injection pumps and the adjustment of the fuel control leverage. It is also needed for priming the fuel-injection pumps in the event of oil pump stoppage. Provision is made for operating the engine protection equipment by hand.

During the starting of the engine, some of the starting air is applied to piston *8* which causes the latch to move downwards with the aid of lever *6*. Once in the downward position, the latch will cut out fuel delivery should the oil pressure appear to be low in the system at starting.

REVIEW QUESTIONS

1. How is engine operation monitored?
2. What engine protection systems are in use?
3. What instruments are employed to monitor engine operation?

Chapter XVIII

TURBOBLOWERS AND SUPERCHARGING

91. Air and Exhaust Ducting

On a normally-aspirated four-stroke diesel engine, the air required for burning the fuel is drawn into a common manifold due to a vacuum set up in the cylinders by the pistons when these are travelling from TDC to BDC. Each cylinder connects to the air manifold by way of a short pipe. A strainer or an air filter is provided at the inlet of the air manifold which is commonly located in the engine room. Also in use are air intakes located at the deck, which are designed to reduce noise and to increase the volumetric efficiency of the engine cylinders. However, an induction trunk, unavoidable in this case, extends the length of ducting and brings about an increase in the resistance to the flow of air. On a supercharged four-stroke diesel engine, the inlet manifold is connected to the air blower.

On a two-stroke diesel engine, air is admitted into the scavenging pump which compresses it and feeds into the air receiver communicating with the cylinders through scavenging ports. Depicted in Fig. 120 is a two-lobe rotary scavenging pump (known as Roots blower) fitted to a two-stroke diesel with uniflow valve scavenging. Two-stroke engines with reciprocating scavenging pumps are provided with silencers which reduce the noise produced by the incoming air. Illustrated in Fig. 121, such a silencer features muffling elements in the form of divergent nozzles.

A reciprocating scavenging pump driven by crank *1* of the engine crankshaft can be seen in Fig. 122. Two pistons *5* and *11* are attached to piston rod *2*. A sucking action created in the spaces above the pistons during their travel from TDC to BDC causes an inflow

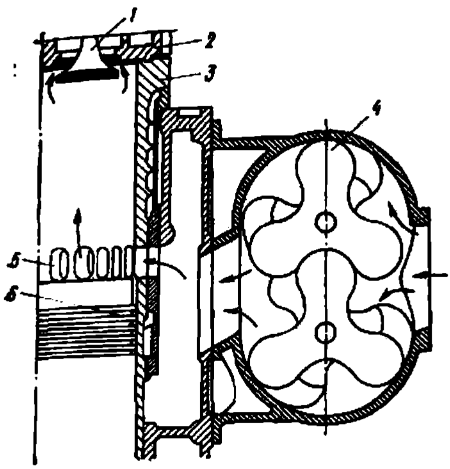


Fig. 120. Two-lobe rotary scavenging pump used on two-stroke diesel engines with uniflow valve scavenging

1—exhaust valve; *2*—cylinder head; *3*—cylinder liner; *4*—rotor; *5*—air receiver; *6*—piston

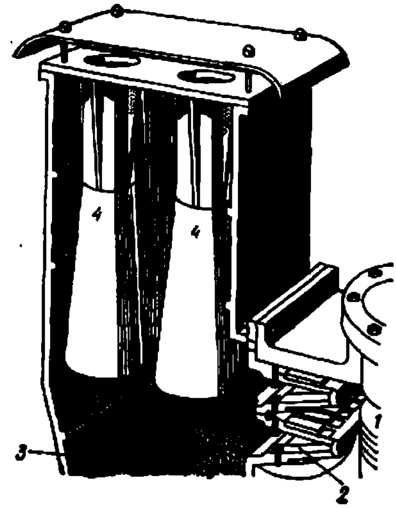


Fig. 121. Silencer with divergent muffling nozzles

1—piston of scavenging pump; *2*—pump valves; *3*—pump casing; *4*—divergent nozzles

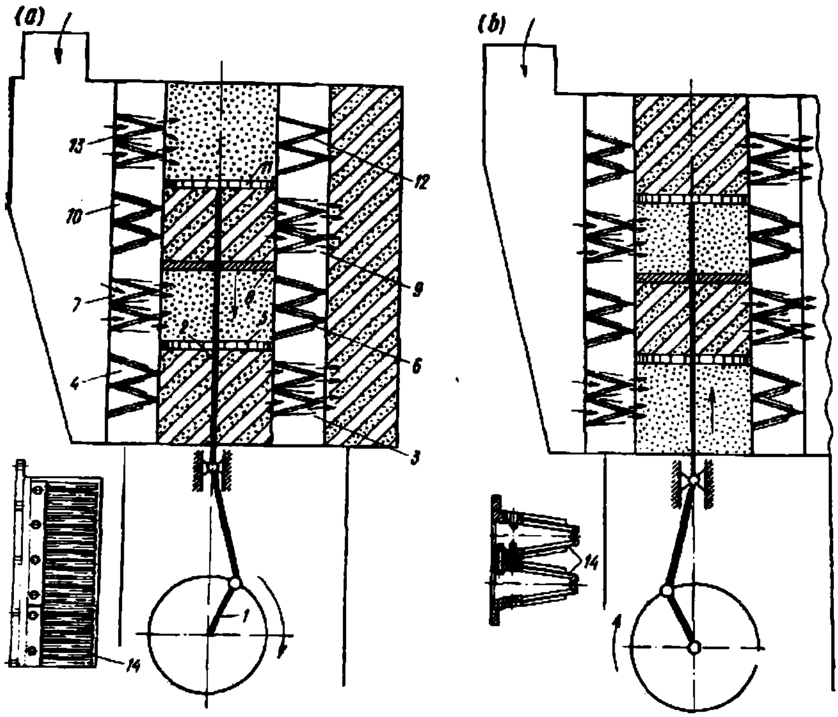


Fig. 122. Schematic of reciprocating scavenging pump in operation

of air into the pump through valve boxes *7* and *13* (Fig. 122*a*). At the same time, air is also discharged from the pump by way of valve boxes *3* and *9*. When the pistons, are travelling in the opposite direction, suction is via valve boxes *4* and *10* and the delivery into the air receiver through valve boxes *6* and *12*. The construction of the valve box *14* is presented in Fig. 112*a* and *b*.

The air receiver is a hollow cast or welded structure intended to minimize undesirable pressure pulsation. A relief valve is commonly

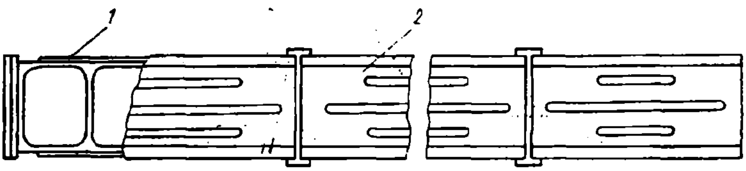


Fig. 123. Exhaust manifold insulated with shields

fitted at the end face of the receiver to safeguard it against excessive air pressure and oil vapour explosion. Inspection holes are provided in the air receiver opposite each cylinder to give access to the piston rings, piston skirts and ports, both scavenging and exhaust ones.

The exhaust manifold of a diesel engine is made up of separate sections of cast or welded construction. On the supercharged diesel engines, the exhaust manifold is attached to a flange of the compressor-driving turbine. To protect the personnel from the hazard of burns, exhaust manifold *1* may be provided with water jackets or shields *2* (Fig. 123). Quite frequently, a waste heat steam generator is connected to the exhaust manifold to recover the heat of the gases.

92. The Supercharging of Diesel Engines

40 000-50 000 kW that the modern marine diesel propulsion plant is capable of developing is considered a maximum power which cannot be exceeded by such traditional means as an increase in the cylinder bore or in the number of cylinders. In fact, the largest engines built abroad have a cylinder bore ranging from 850 to 1 060 mm—a limit beyond which the mass and size of the engine become awkward, and the thermal stresses in the cylinders and pistons, intolerable. Likewise, the maximum practical number of cylinders in an in-line diesel engine is ten to twelve for reasons of simplicity and convenience in operation and repair.

It will be recalled that engine output is a function of such parameters as mean piston speed c_m and mean effective pressure p_e in the cylinder. It has been also proved that the stresses in, and relative deformations of, engine components increase directly with the

mean piston speed raised to the square. This means that a two-fold increase in c_m results in a four-fold stress. To sustain such a stress parts should be made of special materials though their strength is as limited as the cylinder bore and the number of cylinders.

It is also necessary to take into consideration that the mean piston speed is a factor influencing the rate of wear and, consequently, the life of the engine. To keep it within reasonable limits, preference is given to low- and medium-speed diesel engines having a mean piston speed of 8-12 m/s. No diesel engine in use nowadays operates at a mean piston speed over 15-18 m/s.

Thus, the main factor enabling the boosting of modern marine propulsion plants is the mean effective pressure p_e. Since $p_e = p_i \eta_m$, it is obvious that a high mean effective pressure calls for keeping the mechanical losses at minimum. This fact justifies, for example, the tendency to relieve the engine from all auxiliaries mounted thereon, such as scavenging pumps and cooling-water pumps. However, this step will lead only to an insignificant increase in engine power output.

More promising appear to be the steps aimed to increase the value of p_i which is the indicated mean effective pressure. To that end, more fuel is to be burned per cycle which is a problem solvable by increasing the density and mass of the air charge fed into the cylinder without changing the excess air. In other words, the engine output can be boosted by means of supercharging.

Let us consider what supercharging means as a process. Obviously, the more fuel is being injected into the cylinder, the higher is power output produced by the cylinders. However, at some point the fuel fed into the cylinder will fail to burn completely due to a shortage of the air. By increasing the amount of air admitted into the cylinder, it is possible to burn more fuel and, consequently, obtain greater power output. This is the essence of supercharging.

For a quite long period, supercharging was used exclusively on four-strokes, and practical schemes of supercharged two-stroke diesel engines made their appearance in marine application not earlier than some 15 to 20 years ago.

Depending on the way in which the air blower obtains the drive, distinction is made between mechanical supercharging and turbine supercharging.

Mechanical supercharging involves the compression of air from atmospheric to supercharge pressure in blower *1* driven from crankshaft *2* of the diesel engine and the cooling in intercooler *3* (Fig. 124*a*). In this case, some of the indicated engine power is wasted in the drive. This is a factor that markedly decreases the mechanical efficiency of the engine and increases its fuel consumption if the supercharge pressure is in excess of 15.7×10^4 to 16.7×10^4 N/m² (1.6-1.7 kgf/cm²). Therefore, it is practical to use mechanical supercharging on engines requiring a supercharge pressure

between 13.2 × 10^4 and 22.2 × 10^4 N/m^2 (1.5-2.5 kgf/cm^2). At higher pressures, this type is inefficient and finds application mainly as the first or second stage of a combined supercharging system.

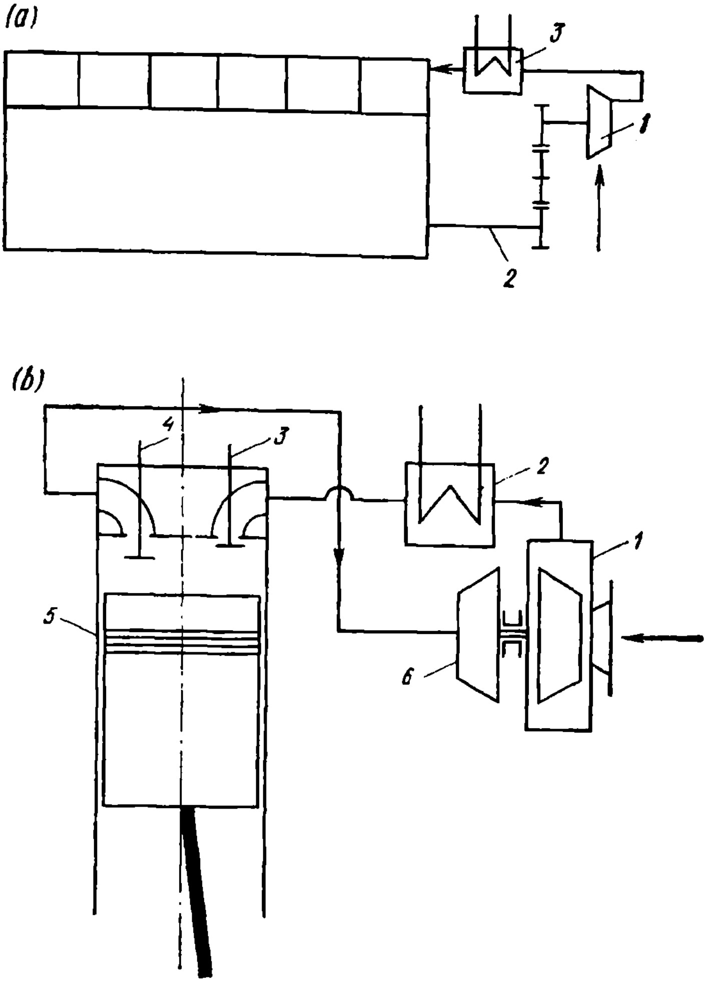


Fig. 124. Schematic of supercharging systems used on diesel engines

Turbine supercharging (Fig. 124*b*) involves the use of exhaust gases which drive the blower with the aid of a turbine, the centrifugal blower and the gas turbine sharing the same shaft. The unit compris-

ing the two machines is referred to as turbocharger which is standard equipment of most modern diesel engines.

Consider joint operation of diesel engine *5* and turbocharger *6* driven by the exhaust gases admitted from the cylinder by way of exhaust valve *4*. Unlike the mechanical supercharging equipment, the turbocharger is not geared to the diesel. The air compressed in compressor *1* to a specified supercharge pressure p_s is fed into the air receiver via intercooler *2*, reaching hence the cylinder through inlet valve *3*. The turbocharger is prone to self-regulation, for with an increase in the load on the engine the amount of the exhaust gases increases and so does the velocity of their impact on the turbine blades with the result that the turbine speed increases. The increase in turbine speed accelerates the compressor. Thus, with an increase in engine load, the amount of air admitted into the cylinder and its pressure automatically increase. The turbine speed is commonly between 6 000 and 30 000 rpm.

The mechanical efficiency of turbocharged diesel engines with air intercoolers is as high as 0.9-0.92 and the specific fuel consumption is as low as 177-190 g/kW h (130-140 g/hp h); the turbocharger provides for an effective pressure of up to 176×10^4 N/m^2 (20 kgf/cm^2). In other words, no other thermal prime mover is capable of operating so economically.

Not uncommon in modern marine practice are turbocharged diesel engines with a specific fuel consumption varying over the range from 204 to 218 g/kW h (150-160 g/hp h). The supercharge pressure is measured after the turbocharger. The ratio of the mean effective pressure with supercharging, p_{es}, and that without supercharging, p_e, is termed supercharge ratio. It varies between 1.2 and 1.8. Higher values invite difficulties, for excessive high mechanical and thermal stresses are set up in an engine with an increase in p_{es}. A practical way of maintaining the thermal stresses at an appropriate level on a supercharged engine is to increase the excess air, i.e. to reduce the mean temperature of the cycle. A similar effect may be obtained by giving the combustion chamber a more thorough scavenging than commonly used on other engines or by intercooling the scavenging air.

The intercoolers in use are of various type, e.g. shell-and-tube ones with round or flat tubes and plain or corrugated cooling fins, plate heat exchangers, etc. Catering for normal operation of the turbocompressor, they increase the mass of air charge per unit volume, rendering it heavier by 2-2.5% with every 10-degree Celsius decrease in temperature. The result is a reduced temperature of the operating cycle and a relaxation of thermal stresses with a simultaneous increase in the supercharge pressure.

The gas turbine may be of the axial-flow or radial-inward-flow type. In the former type, the exhaust gas follows a straight flow

pattern without deviations as this is the case in the latter type, therefore the axial-flow turbines have a higher efficiency than the radial-inward-flow ones. However, an axial-flow turbine must operate under the exact conditions for which it has been designed or otherwise surging will be imminent, rendering it unsuitable for the use.

Centrifugal compressors are a simple, compact and lightweight facility for furnishing the air to supercharge an engine under an appropriate pressure obtained by virtue of the centrifugal forces acting on the air in operation. Their ratio of compression is 3-3.5.

Axial-flow compressors are used to a lesser degree in supercharging applications. The reason is that high-pressure machines of this type consist of a number of stages and it is a problem to arrange them all on the same shaft with the turbine which is commonly a single-stage one due to a small pressure drop.

It has been noted that supercharging effectively clears the engine cylinders from the burnt gases, increases the volumetric efficiency of the cylinders and decreases the temperature of exhaust gases.

With reference to the way of ducting the exhaust, turbines are classified into pulsed and constant-pressure types; quite frequently both types are used on the same engine. Combined supercharging means the use of mechanical and turbine supercharging simultaneously.

In the case of constant-pressure gas flow, the exhaust from all cylinders escapes into a single manifold of a considerable volume which directs the gases to the turbine nozzles usually located at its end.

A plan widely used to increase the effect of supercharging consists in that the exhaust manifold is subdivided into a number of separate sections each connecting to a group of two to four cylinders. The pulsing gas flow from each section is directed to the turbine through a separate inlet connecting to the nozzle assembly. The connection is effected so that the operating cycles in the cylinders connected to the same manifold section are shifted with respect to each other by 240 deg of crankshaft rotation. This arrangement provides for the shifting of the exhaust pressure pulses in time by a maximum amount so as not to interfere with each other.

The supercharging by way of pulsing gas flow puts to a useful purpose not only the heat of exhaust but also the velocity head of the gases escaping from the cylinders (Fig. 125*a*). A sharp change in the pressure inside the exhaust manifold occurs just short of the exhaust stroke. To benefit from the pressure pulses, it is practical to locate the turbine at the shortest possible distance from the exhaust valves or ports and to connect it to each group of cylinders via a manifold of a minimum volume. Quite frequently, this requirement is met by providing a number of turbines which are located close to the engine and by using short lengths of pipes for directing the exhaust

from each cylinder to the nozzle assemblies so that kinetic energy losses of the gases are kept at a minimum. In the wake of each pressure pulse reaching the turbine there is a pressure drop to a point below the scavenging air pressure which facilitates the filling of the cylinder with a fresh air charge.

The supercharging by pulsing gas flow is employed on both two- and four-strokes. However, on low- and medium-output diesel eng-

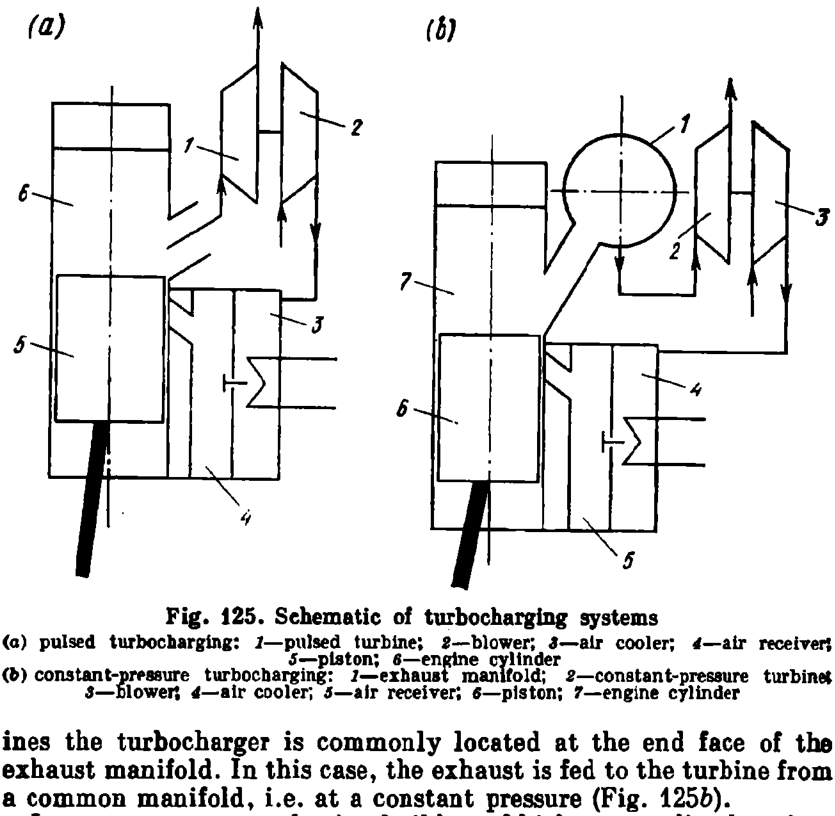


Fig. 125. Schematic of turbocharging systems

(a) pulsed turbocharging: 1—pulsed turbine; 2—blower; 3—air cooler; 4—air receiver; 5—piston; 6—engine cylinder
(b) constant-pressure turbocharging: 1—exhaust manifold; 2—constant-pressure turbine; 3—blower; 4—air cooler; 5—air receiver; 6—piston; 7—engine cylinder

ines the turbocharger is commonly located at the end face of the exhaust manifold. In this case, the exhaust is fed to the turbine from a common manifold, i.e. at a constant pressure (Fig. 125*b*).

In recent years some foreign builders of high-output diesel engines avoid the use of pulsing gas flow and rotary valves in exhaust ports. Their theory is that the supercharging by a constant-pressure gas flow is pivotal to a better filling of the cylinders with air while the thermal stresses remain the same. This practice offers some advantages, such as the use of one turbine only and its discrete location in the system.

In use are also combined supercharging systems comprising a mechanically-driven centrifugal blower and a turbocharger.

93. The Supercharging of Two-Stroke Diesel Engines

Practically all modern two-strokes are supercharged, the air charge being fed into the cylinder under a pressure above the atmospheric one. However, the charging pressure (which is also the scavenging one in this case) is quite moderate, from 10.8 to 14.7 N/m^2 (1.1-1.5 kgf/cm^2) and 18.6 N/m^2 (1.9 kgf/cm^2) on rare occasions. The reason the charging pressure is not carried higher is that too much of the engine power would be required to drive the blower, and too much energy would be lost in the exhaust, since the exhaust pressure increases rapidly with the charging pressure. This situation arises due to the fact that the exhaust gases are evacuated from a two-stroke diesel engine forcibly, due to the scavenging air pressure. As a result, not only some of the air charge is lost through exhaust ports but wasted is also the thermal energy of the gases which mix up with the air. The temperature of exhaust gases is 573-623 K (300-350°C) on two-strokes, and 673-823 K (400-550°C) on four-strokes.

Early attempts to substitute a turbocharger for a mechanically-driven blower have failed due to a lack of power on the part of the turbine at starting the engine and during low-speed ship's maneuvering. A low turbine power output results from a meager supply of exhaust gases when the cylinders operate under own power.

To overcome this handicap and to supply the air in amounts adequate for the starting and low-speed running of the engine, an arrangement has been to provide the engine with a mechanically-driven scavenging blower which functions as an extra stage of supercharging. However, this plan has failed to gain recognition on modern high-output two-strokes. Marine practice gives preference to other supercharging systems, e.g. those of the pulse type offering higher power outputs per cylinder, which have been discussed above.

A point to be noted is that by resorting to a high supercharge pressure alone one cannot adequately boost the engine, for this gives rise to the problem of thermal overstressing of the cylinder walls due to the high temperature resulting from an increase in the pressure. The charge air can be cooled by 20 to 50°C in intercoolers, but they add weight to the engine and require extra attention.

Thus, as far as two-strokes are concerned, the supercharging invites a number of difficulties which can be, however, overcome by employing combined supercharging systems in any of the following ways.

1. A combined two-stage supercharging system employing a turbocharger in conjunction with a scavenging pump geared to the crankshaft, which alternately deliver the air into a common air receiver (Fig. 126) through an intercooler. The scavenging pump functioning as the second stage of supercharging is either a rotary lobar or a

reciprocating one. The displacement effect of the piston underside space is used sometimes in the capacity of the reciprocating pump (Fig. 127).

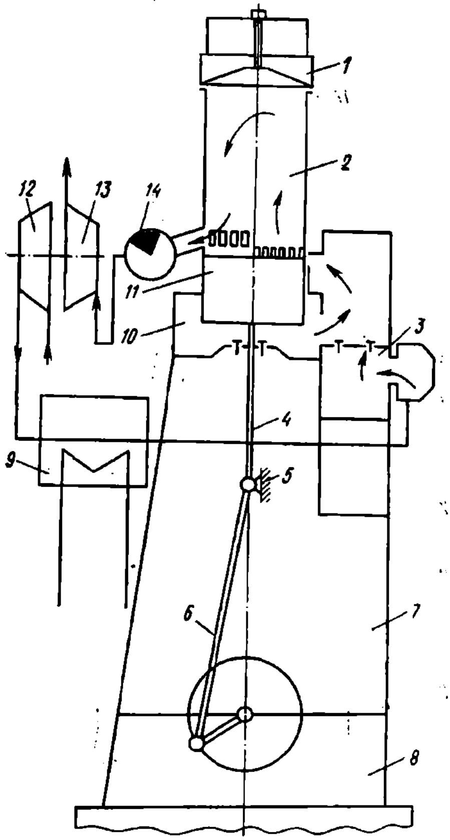


Fig. 126. Schematic of turbocharging system employing displacement effect of piston underside space (successive supercharging)

1—cylinder head; 2—engine cylinder; 3—first-stage air receiver; 4—piston rod; 5—crosshead; 6—connecting rod; 7—engine frame; 8—bedplate; 9—air intercooler; 10—second-stage air receiver; 11—piston; 12—blower; 13—turbine; 14—rotary exhaust valve

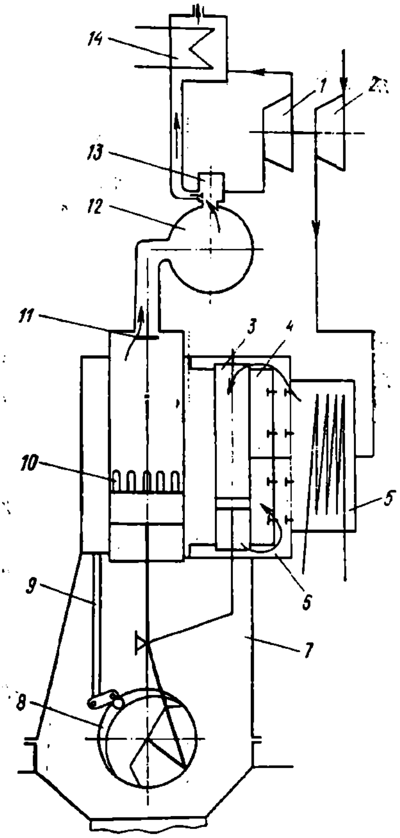


Fig. 127. Schematic of combined two-stage turbocharging system employing turbocharger and scavenging pump

1—turbine; 2—blower; 3—reciprocating scavenging pump; 4, 6—air receivers; 5—air intercooler; 7—engine frame; 8—balance weights; 9—tie rod; 10—scavenging ports; 11—exhaust valve; 12—exhaust manifold; 13—valve box; 14—waste-heat boiler

2. A combined supercharging system involving simultaneous delivery of air into the receiver from a turbocharger, supplying the bulk of the air charge through a cooler, and from the piston under-

side spaces of the cylinders catering for the balance of air; alternatively, a reciprocating or a rotary lobar pump may be used (Fig. 128).

3. A combined two-stage supercharging carried out by a turbocharger (first stage) and engine-driven reciprocating pumps (second

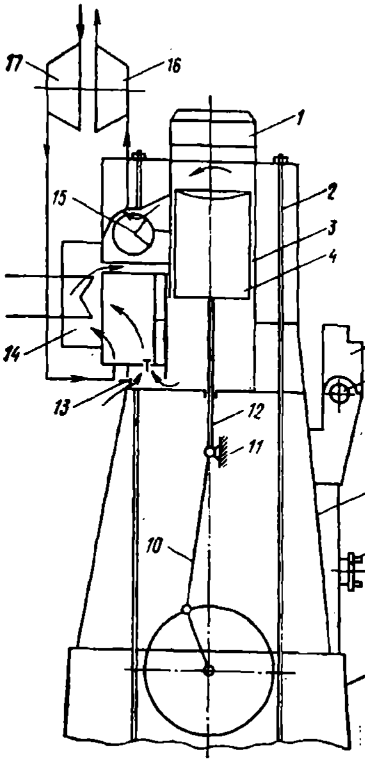


Fig. 128. Schematic of turbocharging system employing displacement effect of piston underside space in conjunction with turbocharger (parallel supercharging)

1—cylinder head; 2—through bolts; 3—engine cylinder; 4—piston; 5—fuel-injection pump; 6—camshaft; 7—engine frame; 8—control hand wheel; 9—bedplate; 10—connecting rod; 11—crosshead; 12—piston rod; 13—valve; 14—air cooler; 15—rotary exhaust valve; 16—turbine; 17—blower

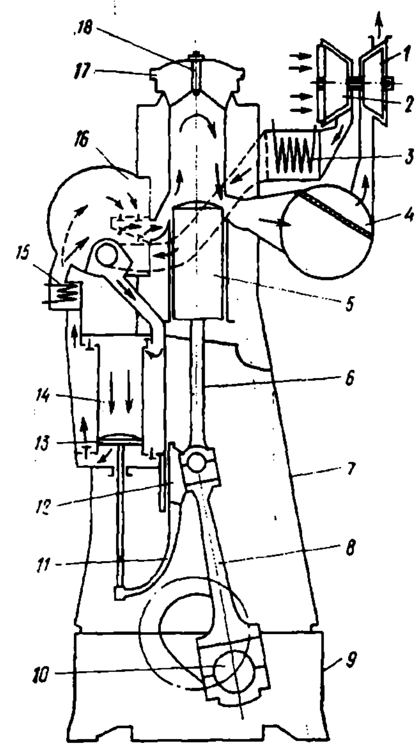


Fig. 129. Schematic of combined two-stage turbocharging system employing turbocharger and engine-driven reciprocating pump

1—turbine; 2—blower; 3, 15—air coolers; 4—exhaust manifold; 5—piston; 6—piston rod; 7—engine frame; 8—connecting rod; 9—bedplate; 10—crankpin; 11—push rod actuating engine-mounted scavenging pump, reciprocating type; 12—crosshead; 13—piston of scavenging pump; 14—scavenging pump; 16—air receiver; 17—cylinder head; 18—fuel injector

stage), with the supercharge ratio controlled automatically depending on engine load (Fig. 129).

4. A combined supercharging system incorporating engine-mounted superchargers arranged to operate in conjunction with the piston

underside spaces of the engine cylinders, some of these spaces delivering the air charge simultaneously with the superchargers, and others—alternately.

94. Some Supercharging Systems Used in Marine Practice

The supercharging of marine diesel engines is the most effective way of boosting their output per cylinder simultaneously with reducing the engine mass and size. Therefore, nearly all diesel engines of low and high output are built predominantly with a turbocharging facility. The supercharging system used in each particular case is adopted depending on the engine design, the number of cylinders, the way of disposing of the exhaust gases, supercharge ratio, etc.

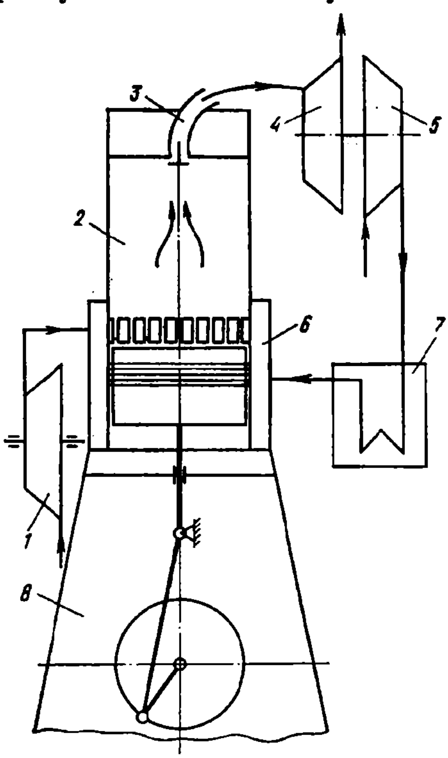


Fig. 130. Schematic of turbocharging system featuring more than one blower

1 5—blowers; 2—engine cylinder; 3—exhaust pipe; 4—turbine; 6—air receiver; 7—air cooler; 8—engine frame

The ever-growing demand for turbochargers has been conducive to setting up a new industry specializing in the production of this equipment. Marketed nowadays are turbochargers suitable for use on diesel engines of any output so that their manufacture has been abandoned by engine builders.

The number of turbochargers, an engine may be fitted with, varies with the number of cylinders. If a pulse system is employed, standard practice is to provide a turbocharger per each three engine cylinders. Therefore, the 7ДКРН 74/160 diesel has two, and the 9ДКРН 50/110 engine, three turbochargers.

Consider some of the supercharging systems used on modern marine diesel engines.

On the Burmeister & Wain engines, air blowers with filters at the inlets deliver air into a common air receiver under a pressure of from 17.6×10^4 to 19.4×10^4 N/m^2 (2.0-2.2 kgf/cm^2) after their

hot outflow having a temperature of 358-363 K (85-90°C) has passed through coolers (Fig. 130). The pulsing exhaust reaches the turbines through the exhaust valves. Simple as it is, this system is reliable in operation and, dispensing with crankshaft-driven scavenging pumps, displays high efficiency (0.90-0.92).

The Sulzer RD engines feature a two-stage turbocharging system with a turbocharger as the first stage, and the piston underside effect catering for the second stage. The air is compressed by the turboblower and delivered through an aftercooler into a common scavenge receiver. From here it is aspirated through non-return plate valves into the piston underside space and delivered into individual scavenging chambers separate for each cylinder. The displacement effect of the piston underside consists in that the air compressed by the piston during the downstroke to $17.6 \times 10^4 = 22.0 \times 10^4$ N/m^2 (2-2.5 kgf/cm^2), which causes the non-return valves that separate the cylinder and aftercooler receivers to close, is delivered into the cylinders through the scavenging ports when these are uncovered by the piston crown. When the piston reverses its travel and moves towards the TDC position, the pressure in the piston underside space decreases and the non-return valves open, admitting air into this space. A rotary valve, chain-driven from the crankshaft, is arranged to close the exhaust duct until the beginning of the exhaust stroke so as to prevent air leaks through the exhaust ports after the piston has covered the scavenging ports. The air pulses from the piston underside spaces assure reliable starting of the engine and prevent blow-by from the cylinder into the receiver. However, some 5% of the rated power are lost to compress the air in the piston underside spaces.

Fiat diesel engines employ a two-stage supercharging, using constant-pressure turbines (see Fig. 129). Turbine-driven centrifugal blowers constitute the first stage and reciprocating scavenging pumps of the double-acting type are functioning as the second stage. The pumps are mounted on the engine at each of the cylinders and obtain the drive from the crosshead by way of a leverage. Intercoolers are provided downstream of the first and second stages, and the outflow from the second stage is directed into a second-stage air receiver. The engine-mounted scavenging pumps provide for easy starting and reliable operation of the engine at all speeds and under any load. They also supply scavenging air enabling the engine to operate at 50% of the rated power output in the event of the turbocharger failure. The drawback of this system is extra attention required by the scavenging pumps and reduced mechanical efficiency of the diesel engine.

The two-stage supercharging system of the K7Z 70/120C MAN engine incorporates two turboblowers (the first stage) which deliver the air into a scavenge receiver through intercoolers. Also in use is

the displacement effect of the piston underside space, this being filled with air during the upstroke of the piston. On the downstroke, the piston compresses the air to a given pressure, and delivers it into a second-stage air receiver. At the same time, the air contained in the first-stage receiver is fed into the double-acting reciprocating pumps of the cylinders to be delivered into the second-stage receiver.

Recent models of the engine dispense with the scavenging pumps which reduce the mechanical efficiency of the engine. The displacement effect of the piston underside space is resorted to in only the 2/3 of the engine cylinders, the air flow thus induced being fed into the scavenge receiver alternately with the flow from the turboblowers. The rest of the spaces draw air from the engine room and deliver it into the receiver in parallel with the main flow. An increase in the scavenging air pressure resulting from an increase in engine output or speed causes to open the non-return plate valves between the first- and second-stage air receivers. Thus, the air flows from one receiver into the other, by-passing the piston underside spaces. No air also enters the spaces from the receiver, for the pressure there is higher than the pressure in the spaces. Consequently, they set about to idle. This arrangement reduces the power requirements for the compressing of air and improves the mechanical efficiency of the engine.

Götaverken diesel engines also employ two-stage supercharging (see Fig. 126). A constant-pressure turbine is actuated by the gases from the exhaust manifold connecting to the cylinders via the valves. Engine-mounted reciprocating pumps of the double-acting type are provided at each cylinder. They are driven by the crosshead through a leverage. The air receiver consists of two parts, an internal and an external one, fitted with non-return valves. The compressed air leaving the compressor is passed through the cooler. When the piston of the scavenging pump is on the downstroke, the valves in the external receiver section open and air enters one of the pump spaces. At the same time, the air contained in the other pump space is delivered into the internal section of the receiver and hence into the engine cylinder. Unlike other two-stage supercharging arrangements, this system enables an automatic increase in air pressure with an increased engine power output.

95. Supercharge Air Cooling

High supercharge pressure increases the temperature of air which leads to excessive thermal stresses in the pistons and cylinders of a diesel engine. For normal operation of supercharged engines, the air is cooled in special heat exchangers. The cooling of air increases the density of air charge and the mean effective pressure, maintaining the thermal stresses within allowable limits.

The supercharge pressure is commonly anywhere between 14.7×10^4 and 24.2×10^4 N/m^2 (1.5-2.5 kgf/cm^2). The rate of cooling a heat exchanger can afford depends on the temperature of incoming air and cooling water, the rate of heat exchange, the velocity of air and water, etc. The requirements a heat exchanger must

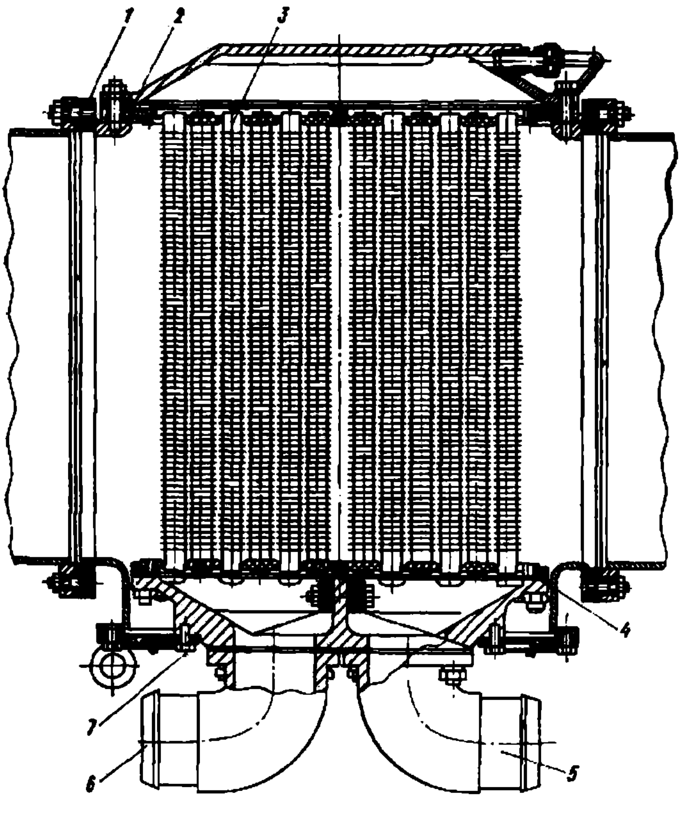


Fig. 131. Heat exchanger

meet are compactness and good heat transfer even in the tropics where the outboard water temperature is 301-305 K (28-32°C). Most of the marine diesel engines require that the air temperature after cooling is not over 323 K (50°C).

A heat exchanger of a type 58 two-stroke diesel engine is depicted in Fig. 131. It consists of welded shell *1* and tube plates *4* running between which are tubes *3* with longitudinal ribs serving to extend the surface of heat exchange between the air and water. Shell covers

2 and *7* provided at the top and bottom of the shell, respectively, form spaces *6* and *5* for feeding and discharging the cooling water. Zinc corrosion inhibitors are provided inside the shell. One of the tube plates is arranged to shift axially, compensating for thermal expansion. Finned tubes are made up each of a core in a nickel-copper alloy wound around which are helical fins of an aluminium alloy. The bimetallic tubes are vibration-resistant, not susceptible to fouling on the inside and, therefore, offering low resistance to the air flow. Displaying a long service, heat exchangers with bimetallic tubing are widely used on many marine propulsion installations.

To improve heat transfer, use is made of plate heat exchangers comprising a series of flat tubes in the form of plates arranged so that the air and cooling water are directed alternately into the passages formed by the plates. Troughs in the plates produce turbulence enhancing the heat transfer to the water. Plate heat exchangers combine high efficiency with low mass and compactness. Their disadvantage is rapid blocking and difficulties experienced in reconditioning a heat exchanger; also in case of a defective plate it is necessary to renew the whole unit.

96. Automatic Supercharge Regulation

As already pointed out, the solution to the problem of further boosting of diesel engines lies in the introduction of gas turbine supercharging. However, difficulties are commonly experienced in selecting the right turbocharger, for there must be known the relationship between the flow rate of air and the ratio of compression the turboblower can afford. The inertia of the rotor renders a turboblower incapable of responding instantaneously to a load change. Sudden load increase results in the shortage of air leading to a smoky exhaust and excessive fuel consumption.

There are a number of means intended to cope with such transients. Depending on the engine speed, the turbocharger performance may be regulated by means of an arrangement depicted in Fig. 132. It comprises servomotor *3*, speed-sensitive spool valve *2* and fuel-delivery adjuster *4*. The supercharge ratio produced by turbocharger *1* can be altered depending on engine speed by appropriately setting the turbine nozzle vanes simultaneously with shifting the fuel-injection pump rack *5*, using the servomotor. Also in use are systems for adjusting turboblower performance depending on engine load.

Recently met with on some of the turbocharged diesel engines is the so-called Hyperbar system. Capable of boosting output without imposing the penalty of high mechanical and thermal stresses, it holds out a special promise as a means of assuring reliability and long service life of diesel engines.

For operation, the system relies on the splitting of the flow of scavenging air leaving turboblower *11* (Fig. 133) into two streams, one fed as usually into the diesel engine over duct *2* and the other by-passed into exhaust manifold *7* by way of duct *4*. The exhaust

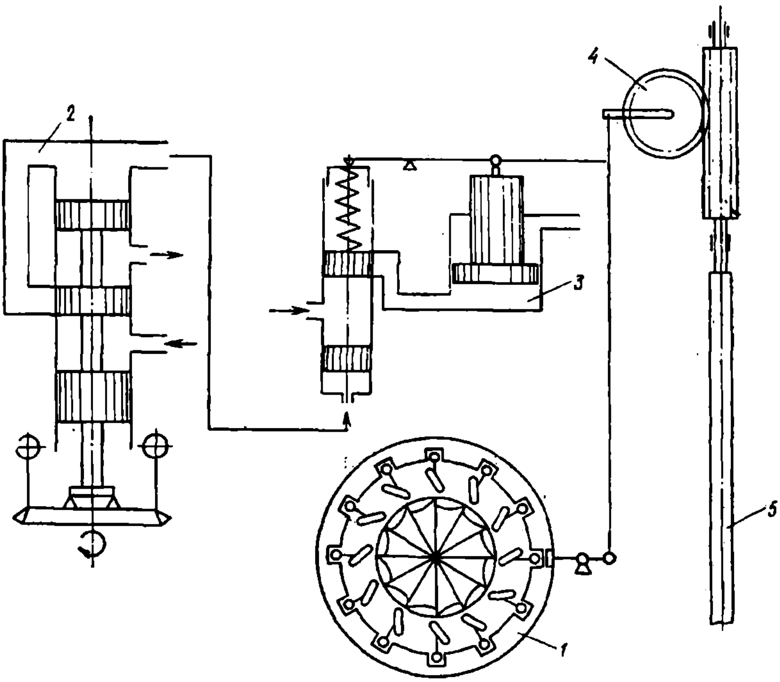


Fig. 132. Schematic of system for regulating turbocharger performance depending on diesel engine speed

manifold contains combustor *8* with an ignitor and connects to device *9* for the control of flame intensity. The air admitted into the exhaust manifold is used to burn fuel fed into the combustor by pump *6*. The resulting gases are ducted to gas turbine *10*. The system also incorporates air cooler *3* and electric motor *1* which sets into motion the turboblower during the starting of the engine. By-pass control valve *5* adjusts the fuel delivery into the combustor so as to set up in the exhaust manifold a gas pressure sufficient to start up the diesel engine. This mode of control assures an increased torque for carrying fractional loads and provides for the performance of the turbocharger which is close to the optimum one. The scavenging air bypass and the combustor enable the starting of the diesel engine at a low com-

pression ratio (6-10) and its operation at a higher mean effective pressure (over 294×10^4 N/m^2 or 30 kgf/cm^2) without a substantial

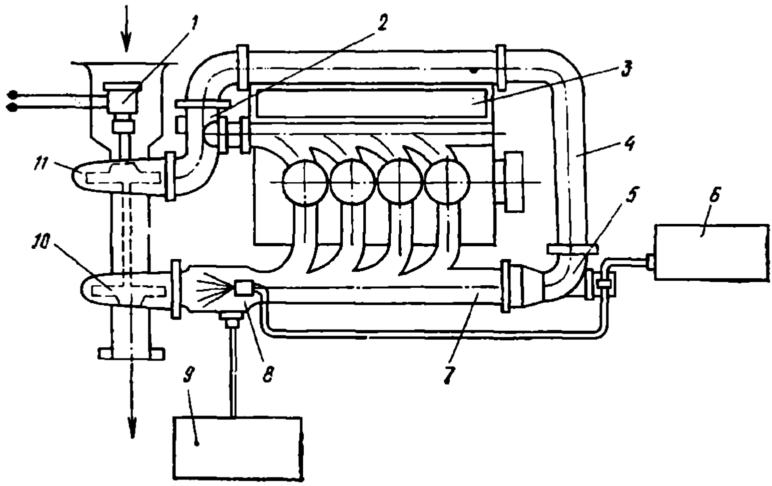


Fig. 133. Schematic of Hyperbar system

increase in the combustion pressure and exhaust temperature. The Hyperbar system can be used on both four- and two-stroke diesel engines.

97. Type TK and TKP Turbochargers

Turbochargers of the TKP series (model TKP-7, TKP-8.5, TKP-11, TKP-14, TKP-18 and TKP-23) feature cantilevered impeller and turbine wheels which are mounted on a shaft supported in internally-arranged bearings. They are used to supercharge high-speed diesel engines.

The supercharging of high-output engines is taken care of predominantly by turbochargers of the TK series (TK-18, TK-23, TK-30, TK-34, TK-38, TK-50 and TK-64) existing in a wide variety of arrangements. On most of the models in use, the bearings are located at the ends of the rotor shaft; alternately, the impeller and turbine wheels can be fitted at the ends of the rotor shaft which are supported in bearings located therebetween. The last-named type is referred to as turbochargers with cantilevered impeller and turbine wheels. They are characterized by a short rotor shaft, compactness and an axially-arranged air inlet contributing to an increase in the blower

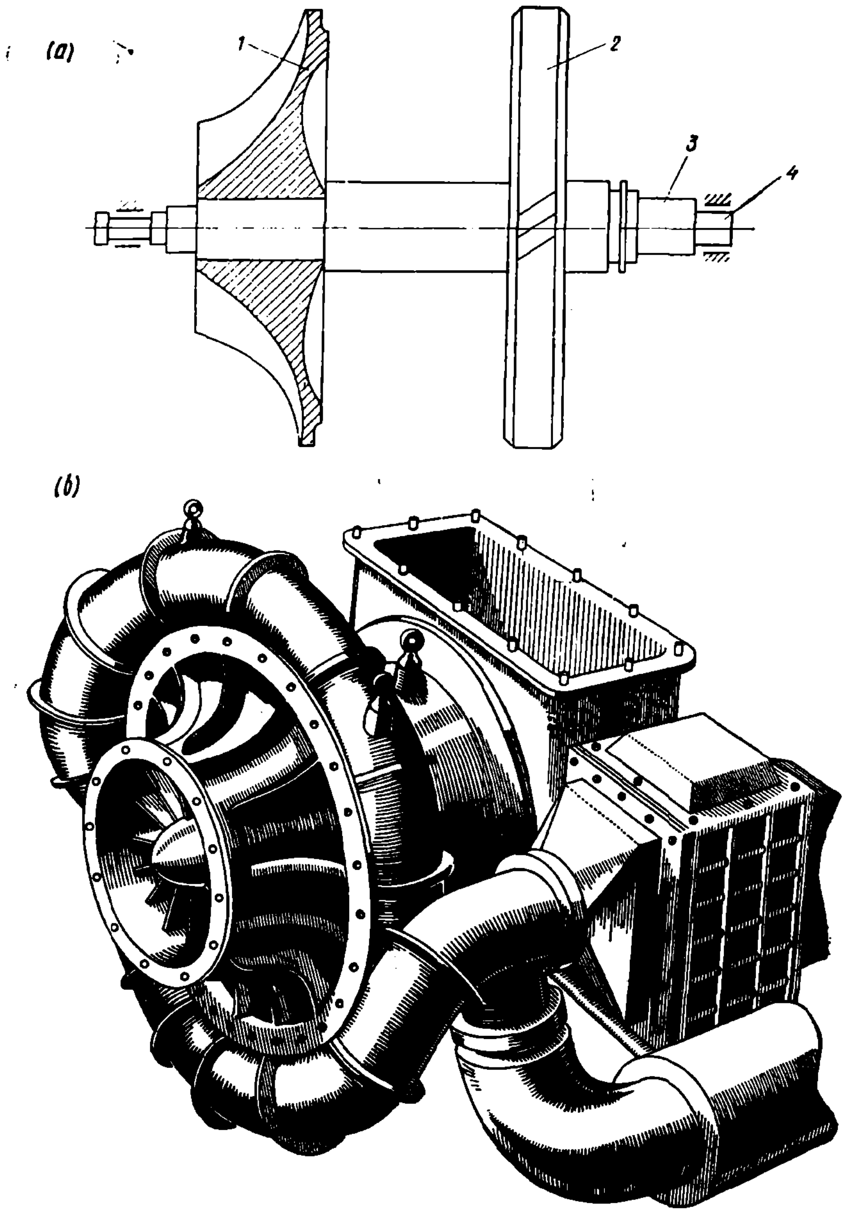


Fig. 134. Type TKP turbocharger

capacity. Blowers are mainly of the centrifugal type, and are simple and reliable in operation.

Turbochargers of the TKP and TK series consist each of rotor shaft *3* carrying compressor impeller *2* and turbine wheel *1*. The shaft is supported by bearings *4* (Fig. 134*a*). A ducted air inlet retains a silencer contributing to a low level of noise emission. A general view of a turbocharger is portrayed in Fig. 134*b*.

Depending on the ratio of compression, turbochargers are classified into three groups which are low-pressure units (ratio of compression, 1.9), medium-pressure installations (ratio of compression, 2.5) and high-pressure machines (ratio of compression, 3.5).

REVIEW QUESTIONS

1. What purpose are the supercharging of diesel engines intended to serve?
2. What is supercharge ratio?
3. What problems arise in supercharging two-stroke diesel engines?
4. Discuss supercharging systems used in marine practice.
5. What purpose does the Hyperbar system serve?

Chapter XIX

BRIEF DESCRIPTION OF MODERN MARINE DIESEL ENGINES

98. Type ЧСП 8.5/11 and ЧСП 9.5/11 Diesel Engines

The ЧСП 8.5/11 and ЧСП 9.5/11 diesel engines are built in this country to propell motor launches and boats. Their major performance data are as follows:

	4ЧСП 8.5/11	6ЧСП 9.5/11
Number of cylinders	4	6
Power, kW (hp)	25 (34)	40.5 (55)
Crankshaft speed, rpm	1 900	1 750
Cylinder bore, mm	85	95
Piston stroke, mm	110	110
Mean effective pressure, N/m^2 (kgf/cm^2)	56.4×10^4-71×10^4 (5.75-7.23)	56.4×10^4-63.8×10^4 (5.75-6.5)
Combustion chamber	turbulence	turbulence
Specific fuel consumption g/kW h (g/hp h)	263-266 (193-195)	272 (200)
Overall dimensions, mm:		
length	1 395	2 590
width	650	585
height	1 000	930
Mass, dry, kg	450	580

The cylinder block/crankcase of these engines is in cast iron and of split construction, the lower part being used as the oil sump. Transverse partitions of the cylinder block give support to the main and crankshaft bearings. The cylinder liners are of the wet type. The underslung one-piece crankshaft works in plain thin-shell bearings, the main journals and crankpins are induction-hardened. The pistons, stamped from an aluminium alloy, are fitted each with three compression and two oil-control rings. The topmost compression ring is chrome-plated. On some modifications of the engines, the pistons are cooled by the oil fed from the main bearings via drillings in the shanks of connecting rods. The hollow piston pins are of the floating type. The stamped I-section connecting rods are split obliquely at the big ends. The cast iron cylinder heads overlay each two cylinder bores and are fitted with an inlet valve, an exhaust valve and a turbulence chamber per bore. The valve-timing gear consists of tappets, push-rods and rocker arms obtaining motion from the camshaft.

The constant-speed governor is a direct-acting, centrifugal one. The fuel system comprises a booster pump, a filter, fuel-injection pumps, injectors and fuel lines. The lubricating system incorporates a strainer, a gear-type pump, primary and secondary filters, a relief valve.

The cylinder liners are cooled by the thermosiphon system and the cylinder heads are cooled by forced circulation. Air is admitted into the engine through an inertia cleaner with an oil bath. For starting, use is made of a hand crank or starter motor. The starting aids include a compression-release gear, heater plugs and a heater. An alarm and monitoring system and engine protection equipment are provided on all engines.

99. Type Ч 15/18 and ЧН 15/18 (3Д-6 and 3Д-12) Diesel Engines

Finding application in marine practice as main and auxiliary engines, these USSR-built diesels (Fig. 135) are characterized by the following performance data:

	Ч 15/18	ЧН 15/18
Number of cylinders	6	12
Cylinder arrangement	in-line	Vee
Power, kW (hp)	110 (150)	173 (235)
Crankshaft speed, rpm	1 000-1 500	1 250-2 000
Cylinder bore, mm	150	150
Piston stroke, mm	180	180
Mean effective pressure, N/m^2 (kgf/cm^2)	31×10^4-76.6×10^4 (3.15-7.82)	45.5×10^4-76.6×10^4 (4.64-7.82)

Specific fuel consumption (with turbocharging), g/kW h (g/hp h)	218-251 (160-185)	204-242 (150-178)
Overall dimensions (reversing and reduction gear included), mm:		
length	2 462	2 464
width	886	1 052
height	1 165	1 210
Mass, dry, kg	1 500-1 750	1 800

The crankcase in cast iron or aluminium alloy is of the split type. The cylinder block, a casting made of the same material as the crankcase, is fitted with wet liners in chromium-molybdenum steel having their bores nitrided. The aluminium alloy cylinder head overlays all bores and is provided with two exhaust and two inlet valves along with an injector per each cylinder. An overhead camshaft is employed to actuate the valves. The drop-forged connecting rods of the I-section are made from alloy steel. A bronze bushing is press-fitted into each small end, and the big ends are split. On the 3Д-12 diesel, the big end of each master connecting rod is provided with a lug for connecting to it the articulated connecting rod.

The drop-forged crankshaft in chromium-nickel-tungsten steel is bored hollow. On the 3Д-12 diesel, each crankpin is provided with a pin connected whereto are the master and articulated connecting rods. The leaded bronze-lined shells of the main bearings are interchangeable.

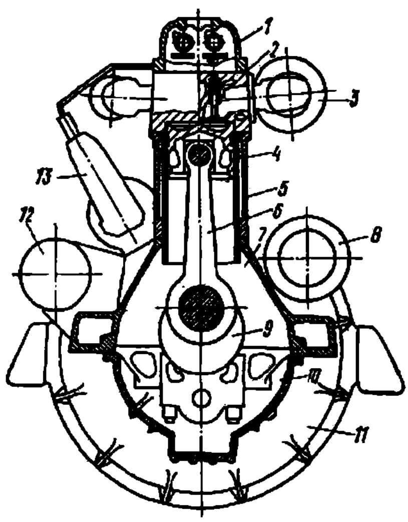


Fig. 135. Type Ч 15/18 and ЧН 15/18 (3Д-6 and 3Д-12) diesel engine

1—camshaft; *2*—cylinder head; *3*—exhaust manifold; *4*—piston; *5*—cylinder liner; *6*—connecting rod; *7*—crankcase; *8*—electric starter motor; *9*—crankshaft; *10*—oil sump; *11*—flywheel casing; *12*—battery-charging generator; *13*—fuel-injection pump

The variable-speed governor is of the pilot-actuated type. The fuel-injection pump is of the multiple-unit reciprocating type. Fuel regulation is effected by rotating the plungers about their axes. The fuel booster pump is a rotary one with guided vanes producing a delivery head of $4.9 \times 10^4 = 6.85 \times 10^4$ N/m^2 (0.5-0.7 kgf/cm^2). The valve-closed injectors operate at a valve-opening pressure of $2\,060 \times 10^4$ N/m^2 (210 kgf/cm^2).

The fuel system incorporates a strainer and an edge-type primary filter and a felt-type fine filter. Some engine modifications make use of fine oil filters with Narva filtering elements and type МЦ-1 centrifugal oil separators.

For cooling, use is made of a closed-circuit cooling system with coolers in which fresh jacket water is cooled by outside water. Circulation is induced by a centrifugal water pump. The supercharging is taken care of by a type TKP-14H turbocharger, and on the 3Д-12 diesel two turbochargers are used. Routine starting is accomplished by a battery-operated starter motor, the stand-by starting is by compressed air under a minimum pressure of 880×10^4 N/m^2 (90 kgf/cm^2).

100. Type 42 ЧНП 16/17, 42 ЧНСП 16/17 and 56 ЧНСП 16/17 Diesel Engines

These diesel engines with a cylinder bore-to-piston stroke ratio of 160/170 mm are built in the USSR for marine application as main propulsion plants. Their performance data are as follows:

	42ЧНП 16/17		56ЧНСП 16/17
Number of cylinders	42		56
Cylinder arrangement		radial	
Power, kW (hp)	1 840 (2 500)		3 680 (5 000)
Crankshaft speed, rpm	1 780		2 000
Cylinder bore, mm		160	
Piston stroke, mm		170	
Mean effective pressure, N/m^2 (kgf/cm^2)	84.4×10^4 (8.6)		113×10^4 (11.5)
Specific fuel consumption, g/kW h (g/hp h)	225 (165)		232 (170)
Overall dimensions, mm:			
length	3 700		3 900
width		1 560	
height		1 630	
Mass, dry, kg	5 450		5 700

Fitted to the barrel-type crankcase are seven cylinder blocks, referred to as monoblocks, which are aluminium alloy castings each comprising the water jacket and the cylinder head for six or eight cylinders depending on the engine modification. The cylinder liners are press-fitted into the blocks. The pistons are aluminium presswork each fitted with two compression rings and two oil scraper rings. The piston pins are hollow and of the floating type. The connecting rod assembly of each block consists of one master connecting rod and six articulated connecting rods. The steel crankshaft is supported in roller bearings. The camshaft is driven by the crankshaft through a vertically arranged gear train.

The variable-speed governor is a pilot-actuated one. The fuel-

injection pump is of the multiple-unit type. Lubrication is taken care of by the dry crankcase, and the cooling system employs the closed-circuit principle. Each diesel engine is provided with a single-stage centrifugal compressor. The starting is by compressed air, and reversing, by a reverse clutch.

101. Type 12ЧН 18/20 (М50Ф-2) Diesel Engine

The 12ЧН 18/20 diesel engine finds application as the main engine of the hydrofoil ships of the "Strela"-, "Kometa"- and "Vikhr-" class. Its leading particulars are:

Number of cylinders	12
Cylinder arrangement	Vee, at 60 deg
Power, kW (hp)	442-736 (600-1 000)
Crankshaft speed, rpm	1 400-1 550
Cylinder bore, mm	180
Piston stroke, mm	200
Mean effective pressure, N/m² (kgf/cm²)	70.6×10^4-94×10^4 (7.21-9.6)
Specific fuel consumption g/kW h, (g/hp h)	212-232 (156-170)
Overall dimensions (reversing and reduction gear included), mm:	
length	2 720
width	1 260
height	1 250
Mass, dry, kg	2 000

The aluminium-alloy crankcase is of the split type, the upper half taking the forces set up by the combustion pressure, and the lower functioning as the oil sump. A scavenging oil pump is contained in the sump. The crankcase gives support to the two cylinder blocks containing the liners which are nitrided on the inside and chrome-plated on the outside.

The pistons are stamped from an aluminium alloy and fitted each with two chrome-plated compression rings and three oil-scraper rings. The piston pins are of the floating type.

The connecting rod assembly comprises six master connecting rods and six articulated ones, all of the drop-forged I-section and in steel. The big ends of the master connecting rods are split obliquely, and each articulated connecting rod is linked to the master rod by means of a pin. The small-end bearings are leaded bronze bushings press-fitted into their places.

The crankshaft is drop-forged as one-piece unit and bored hollow. The crank angle is 120 deg. The main bearings are pressure-lubricated, and the small-end bearings are splash-lubricated. So are the cylinder liners.

The cylinder heads are in an aluminium alloy and each fitted with two inlet valves, two exhaust valves, an injector and a starting valve.

The valve-timing gear of each cylinder block incorporates two crankshaft-driven camshafts. The pilot-actuated variable-speed governor is provided with a PI-device in the form of an oil-filled dashpot.

The fuel system comprises a gear-type fuel booster pump, filters, a fuel-injection pump, injectors and fuel lines. The fuel-injection pump of the multiple-unit type features plunger-controlled ports. The plunger diameter is 13 mm and the plunger stroke is 12 mm. The delivery pressure is 6 850 × 10^4 = 9 800 × 10^4 N/m^2 (700-1 000 kgf/cm^2), and the valve-opening pressure of the injectors is 1961 × 10^4 N/m^2 (200 kgf/cm^2).

Lubrication is taken care of by a circulation lubricating system that incorporates a scavenging oil pump, a pressure oil pump (both of the gear type) and oil filters.

The cooling is by a closed-circuit system. The means of supercharging consists of either one TKP-23 turbocharger or two TK-18H units.

For starting, use is made of the air supplied under the pressure of 735 × 10^4 = 1 470 × 10^4 N/m^2 (75-150 kgf/cm^2); alternatively, an electric starter motor may be employed.

The main engines of the 12ЧCH 18/20 type are commonly provided with a reversing gear or a reduction gear with a speed ratio of 4.15 : 1.

102. Type Ч 18/22 and ЧН 18/22 Diesel Engines

These USSR-built diesels are used as main engines and are characterized by the following performance data:

	6Ч 18/22	6ЧН 18/22	8ЧН 18/22
Number of cylinders	6	6	8
Power, kW (hp)	110 (150)	166 (225)	232 (315)
Crankshaft speed, rpm	750	750	750
Cylinder bore, mm	180	180	180
Piston stroke, mm	220	220	220
Mean effective pressure, N/m^2 (kgf/cm^2)	53×10^4 (5.4)	79.5×10^4 (8.1)	83.4×10^4 (8.5)
Specific fuel consumption, g/kW h (g/hp h)	220 (162)	215 (158)	217 (159)
Overall dimensions (reversing and reduction gear included), mm:			
length	3 213	3 213	4 054
width	1 000	1 000	1 100
height	1 525	1 520	1 853
Mass, dry, kg	4 330	4 330	5 780

The engine structure comprises a bedplate, a cylinder block and cylinder heads. The bedplate is in cast iron and its lower part serves as the oil sump. Its transverse beams are provided with main bearing seats.

The cylinder block is also made of cast iron, and the material of the liners is alloyed cast iron. The cast-iron pistons are provided each with four compression rings and four oil scraper rings fitted pairwise into each groove. The piston pins are of the floating type.

The connecting rods are of the drop-forged I-section. The big-end bearings are provided with inserts in an aluminium alloy containing also antimony and magnesium. The steel main bearing inserts are babbitt-lined.

The crankshaft is drop-forged from a carbon steel and its journals and crankpins are induction hardened. The cams serving to actuate the inlet and exhaust valves are made integrally with the camshaft. In addition to the self-actuated variable-speed governor with a variable feedback effected by a dash-pot, each diesel is fitted with a maximum-speed governor arranged to cut off fuel delivery at a crankshaft speed between 870 and 920 rpm.

The fuel system incorporates a fuel booster pump, a filter, a fuel-injection pump, injectors and fuel lines.

The lubricating system consists of a gear-type oil pump, primary and secondary filters, an oil cooler and an oil-circulating pump used preparatory to starting.

The cooling system is a closed-circuit one. The turbocharger is of the TKP-14 type. The starting is by compressed air or an electric starter motor.

The 8ЧНСП 18/22 diesel engines are equipped each with a reduction and reversing gear with a speed ratio of 1.72 : 1 to 2.46 : 1 in running ahead and the 6ЧНСП 18/22 models employ each a reduction and reversing gear having a speed ratio of 1 : 1 to 2.14 : 1 also for running ahead.

103. Type ДН 23/30 and ДРН 23/30 Diesel Engines

The USSR-built ДН 23/30 and ДРН 23/30 diesel engines used as main propulsion installations are characterized by the following performance data:

Builder's designation	40ДМ	10Д-40
Cylinder arrangement	Vee, at 45 deg	
Number of cylinders	12	16
Power, kW (hp)	1 620 (2 200)	1 800 (2 440)
Crankshaft speed, rpm	750	680
Cylinder bore, mm	230	230
Stroke of piston linked with:		

master connecting rod, mm	300	300
articulated connecting rod, mm	304.3	304.3
Mean effective pressure, N/m² (kgf/cm²)	79.5×10^4 (8.1)	89.2×10^4 (9.1)
Specific fuel consumption g/kW h (g/hp h)	218 (160)	245 (180)
Overall dimensions, mm:		
length	3 924	4 533
width	1 730	1 830
height	2 302	2 863
Mass, dry, kg	9 922	14 600

The cylinder blocks are of welded construction. The drop-forged crankshaft is of the underslug type with nitrided journals and crank pins. A vibration absorber is fitted to the fore end of the crankshaft. The main bearings with leaded bronze-lined thin-shelled inserts are attached to the lower parts of the cylinder blocks. The cylinder heads are of the built-up type, the material of the lower deck being high-duty cast iron and that of the upper deck, an aluminium alloy. Each cylinder head is provided with four exhaust valves, an injector and an indicator cock with a starting valve. The cylinder liners are made from cast iron. The pistons are of the built-up type. Their cooling is effected by the oil fed through the drillings in the connecting rods. Each piston is fitted with four compression rings and two oil-scraper rings. An insert is provided inside each piston. The piston pins are of the floating type. The steel connecting rods are of the I-section. The camshaft is located between the cylinder blocks.

The variable-speed governor is a pilot-actuated one. The fuel system consists of a booster pump, primary and secondary filters, a multiple-unit fuel-injection pump with plunger-controlled ports, injectors and high-pressure fuel lines.

The lubricating system incorporates a crankshaft-driven oil pump inducing oil circulation, primary and secondary filters and an oil cooler. The supercharging is a two-stage one, the first step consisting of two turbochargers operating in parallel and the second step being provided in the form of a centrifugal blower geared to the crankshaft.

The 10Д-40 engine is remotely controlled and equipped with a fluid coupling and a reduction gear providing for a speed ratio of 3.94 : 1 in running ahead. For starting the engines, use is made of compressed air.

104. Type Ч 25/34 and ЧН 25/34 Diesel Engines

Finding application in marine practice as main or auxiliary engines, these models (Fig. 136) are characterized by the following performance data:

	6Ч 25/34	6ЧН 25/34	8ЧН 25/34
Number of cylinders	6	6	8
Power, kW (hp)	221 (300)	331 (450)	589 (800)
Crankshaft speed, rpm	500	500	500
Cylinder bore, mm	250	250	250
Piston stroke, mm	340	340	340
Mean effective pressure, N/m^2 (kgf/cm^2)	52.5×10^4 (5.35)	79.5×10^4 (8.1)	97×10^4 (9.9)
Specific fuel consumption, g/kW h (g/hp h)	222-226 (163-166)	211 (155)	211 (155)
Overall dimensions, mm:			
length	3 970	4 357	5 415
width	1 460	1 460	1 460
height	2 392	2 476	2 655
Mass, dry, kg	10 200	10 710	13 120

The bedplates and cylinder blocks of the engines are in cast iron. The main bearing seats are provided in the bedplate, and the main bearing shells are babbitt-lined. The material of the cylinder liners and pistons is also cast iron. The piston heads are of the concave configuration, and each piston is provided with six rings, two of them being oil-scraper rings. The piston pins are of the floating type. Each cylinder is provided with a separate head and a separate fuel-injection pump.

The steel connecting rods with I-section shanks are drop-forged. A bronze bushing is press-fitted into the small end and the big end is split. The crankshaft is drop-forged and then bored hollow. The camshaft is provided with cams to actuate the inlet and exhaust valves; another set of symmetrically-shaped cams serves to actuate the fuel-injection pumps. The centrifugal variable-speed governor is a self-actuating one. For fuel regulation, use is made of the timed by-pass principle. The lubricating system incorporates a duplex gear-type oil pump, two primary filters of the wire-gauze type and a fine filter with filtering elements. The cooling system is a closed-circuit one. The turbocharger is of the TK-23 type, and for starting use is made of the air under a pressure of 176×10^4-294×10^4 N/m^2 (18-30 kgf/cm^2).

105. Type 20.7/2 × 25.4 (3Д-100) Diesel Engine

The diesel engines of the Д-100 series (Fig. 137) are used on generating sets providing power for propulsive purpose. The engine and

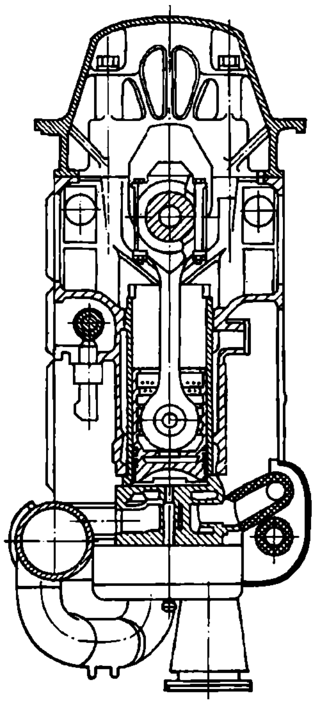


Fig. 136. Type Ч 25/34 and ЧН 25/34 diesel engine

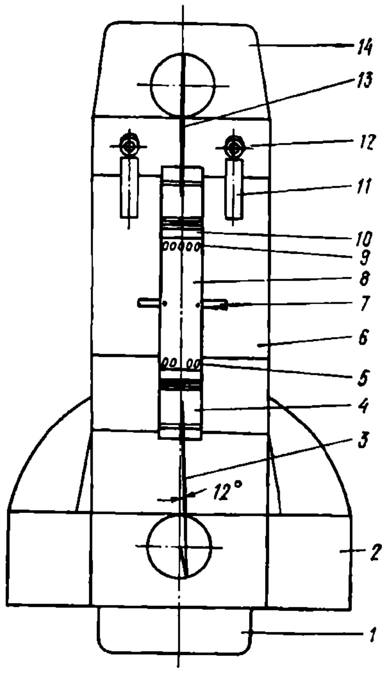


Fig. 137. Type 10ДП and 10ДПН 20.7/2×25.4 opposed-piston diesel engine

1—oil sump; 2—bedplate; 3—lower connecting rod; 4—exhaust port-controlling piston; 5—exhaust ports; 6—engine frame; 7—fuel injector; 8—cylinder liner; 9—scavenging ports; 10—scavenging port-controlling piston; 11—fuel-injection pump; 12—camshaft; 13—connecting rod; 14—top cover

generator are commonly fitted to the same bedplate. The leading particulars of the engines are as follows:

Number of cylinders	10
Power, kW (hp)	1 250 (1 700)
Crankshaft speed, rpm	810
Cylinder bore, mm	207
Piston stroke, mm	2×254
Mean effective pressure, N/m² (kgf/cm²)	60.8×10^4 (6.2)

Specific fuel consumption, g/kW h (g/hp h)	232 (170)
Overall dimensions, mm:	
length over exciter	6 950
width	1 855
height	4 250
Mass, dry, kg	28 3C0

The engine frame is of welded construction with the oil sump at the bottom and a cover at the top. The engine is a two-stroke with uniflow scavenging and fine fuel atomization. The two pistons opposing one another in each of the cylinders during the compression stroke form a common combustion chamber at the midlength of the cylinder liner. The upper pistons impart rotary motion to the upper crankshaft and cover and uncover the scavenging ports. The lower pistons set into motion the lower crankshaft and control the operation of the exhaust ports. Since the lower crankshaft rotates 12 deg in advance of the upper one, it accounts for 70% of the engine power output, the balance being supplied by the upper crankshaft. The cylinder liners are in cast iron, but at the midlength of each of them, i.e. in the zone of combustion, there is press-fitted a steel ring with two injectors, an indicator cock and a starting valve. The pistons are made from cast iron and are cooled by oil. Each of them carries four compression rings and three oil-scraper rings. The piston pins are of the floating type. The connecting rods are steel drop-forgings of the I-section. The big end bearings have bronze shells lined with babbitt. The main bearings are fitted with similar shells.

The crankshafts are castings made of a heavy-duty cast iron. The torque produced by the upper crankshaft is transmitted via a vertical gear train.

The fuel-injection pumps are actuated by two camshafts located at either side of the engine. The centrifugal variable-speed governor is of the pilot-actuated type with a PI-device.

The fuel system comprises a crankshaft-driven booster pump, an edge-type primary filter, a fine filter with felt or paper filtering elements and fuel-injection pumps with plunger-controlled ports.

The lubricating system incorporates an oil-circulating pump, edge-type primary filters, fine filters with replaceable filtering elements, a centrifugal separator and an oil cooler.

The closed-loop cooling system consists of a pump, fresh-water and oil coolers and thermostats which automatically control the fresh-water and oil temperatures.

The two-stage supercharging system comprises two TK-34 turbochargers operating in parallel and a mechanically-driven centrifugal pump or a two-lobe rotary blower. All engines are provided with alarm and monitoring systems.

106. Type ДПН 23/2 × 30 and ДРПН 23/2 × 30 Diesel Engines

These single-acting trunk two-strokes of the double-row cylinder arrangement and with oppositely moving pistons are used as shipboard main engines (Fig. 138). Their leading particulars are as follows:

Builder's designation	61Б, 61В	58А	58В
Number of cylinders	16	16	16
Power, kW (hp)	4 400 (6 000)	3 320 (4 500)	3 320 (4 500)
Crankshaft speed, rpm	850	643	775
Cylinder bore, mm	230	230	230
Piston stroke, mm	300	300	300
Mean effective pressure N/m² (kgf/cm²)	78.5×10^4 (8.0)	77.5×10^4 (7.9)	63.7×10^4 (6.5)
Specific fuel consumption g/kW h (g/hp h)	228 (168)	228 (168)	228 (168)
Overall dimensions, mm:			
length	6 770	6 770	6 923
width	1 900	1 900	1 900
height	3 315	3 315	3 315
Mass, dry, kg	40 000	40 000	40 000

The steel frame of the engine is of a welded construction. The main bearing seats are welded to transverse members of the frame. The main bearing shells are lined with leaded bronze. A cover surmounts the frame. The cylinder liners are in cast iron. A steel ring is fitted at the midlength of each cylinder, and attached to the ring are the starting valve, injectors and indicator cock. The one-piece pistons are also in cast iron of the heavy-duty type. The piston heads are chrome-plated and an insert is provided in each piston. Four compression rings and two oil-scraper rings are fitted to each piston. The crankpin bearings have steel shells lined with leaded bronze. The crankshafts are drop-forged from alloy steel, the main journals and crankpins are nitrided. A gear train is provided to transmit the torque from the four crankshafts to the output shaft.

The drive of the engine auxiliaries (oil pump, fresh and outside water pumps) is accommodated in a welded casing fitted to the front end of the engine frame. The speed governor, starting air distributor and other small auxiliaries obtain the drive from the output shaft. Two camshafts are employed to actuate the fuel-injection pumps located horizontally in the upper part of the engine frame. The engines are provided each with a centrifugal variable-speed governor of the pilot-actuated type and a self-actuating overspeed governor. The fuel system consists of the fuel-injection pumps, a booster pump, filters and injectors. The lubricating system incorporates a duplex

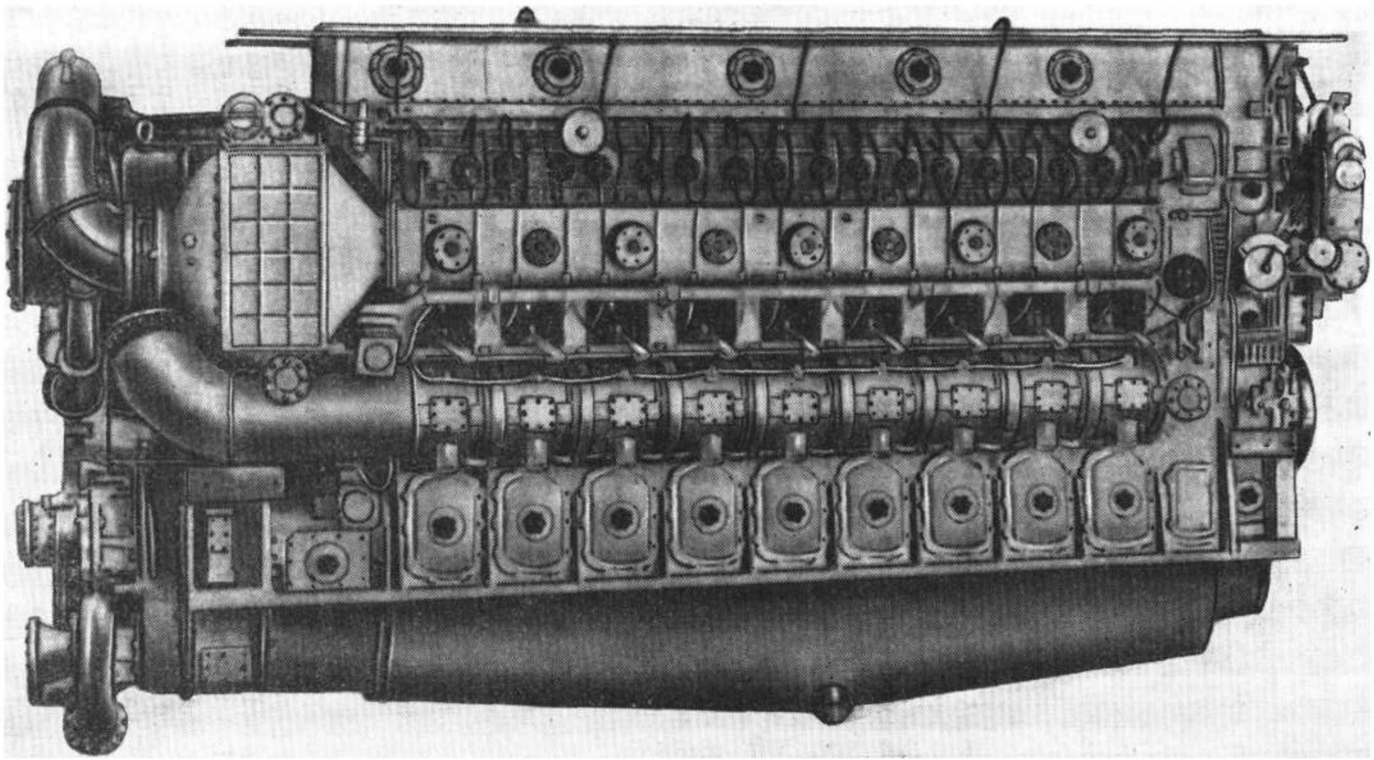


Fig. 138. General view of type ДПН and ДРПН $23/2 \times 30$ diesel engine

oil pump with the suction and the discharge sections, wire gauze filters, and an oil cooler. A thermostat is provided in the main distributing line of the system. The oil sump is attached to the lower part of the engine frame. The cooling system consists of an internal closed-circuit fresh-water setup and an external open-circuit arrangement circulating through which is the outboard water. The fresh-water circuit is provided with a thermostat. The supercharging is taken care of by a single-stage turbocharger and an air cooler.

Each diesel engine is provided with a remote control system and a monitoring and alarm facility. The temperature of jacket water and that of oil are controlled automatically.

107. Type Д and ДР 30/50 Diesel Engines

Finding application in marine practice as main or auxiliary engines, this type (Fig. 139) is characterized by the following performance data:

	4Д; 4ДР 30/50	6Д; 6ДР 30/50	8ДР 30/50
Number of cylinders	4	6	8
Power, kW (hp)	294 (400)	442 (600)	588 (800)
Crankshaft speed, rpm	300	300	300
Cylinder bore, mm	300	300	300
Piston stroke, mm	500	500	500
Mean effective pressure, N/m^2 (kgf/cm^2)	41.7×10^4 (4.25)	41.7×10^4 (4.25)	41.7×10^4 (4.25)
Specific fuel consumption, g/kW h (g/hp h)	242 (178)	242 (178)	242 (178)
Overall dimensions, mm:			
length	3 590	4 405	5 592
width	1 700	1 700	1 700
height	3 140	3 150	3 150
Mass, dry, kg	15 000	18 000	23 000

The engine structure consists of the bedplate, cylinder block and cylinder heads. The welded oil sump is attached to the lower flange of the bedplate. The main bearings have steel babbitt-lined shells. The cylinder block material is a heavy-duty cast iron, cast iron is used for the cylinder liners, too. Each liner is provided with five exhaust and ten scavenging ports. The crankshaft is a one-piece drop-forging. The scavenging pump is arranged to obtain the drive from a crank. The connecting rods are in steel and have shanks of a round section. A bronze bushing is press-fitted into the small end of each connecting rod. The big end is split and both its babbitt-lined halves are attached to the shank by two bearing bolts. The cast iron one-

piece pistons are oil cooled. The oil fed from the main distributing passage to lubricate the main bearings, flows then to the crankpin bearings over drillings in the crankshaft and, finally, reaches the pistons through the drilling in each shank of connecting rod and the slider, cooling down the piston head. The oil is returned to the sump through the hollow bosses of the piston skirt.

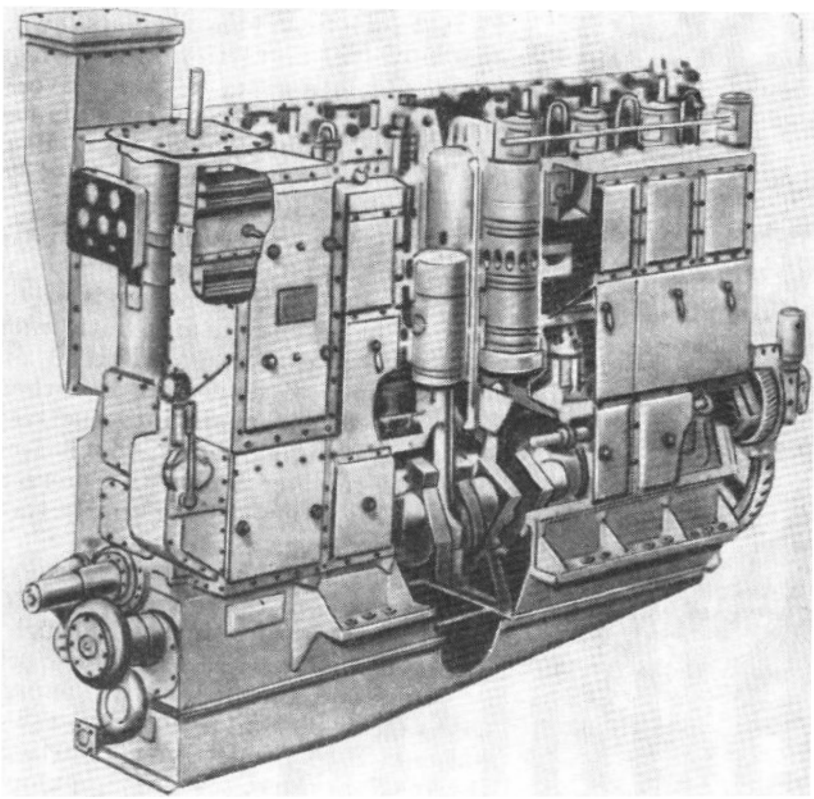


Fig. 139. General view of series Д and ДР 30/50 diesel engine

The steel camshaft is supported in bronze bearings and is fitted with cams of symmetrical configuration, each consisting of two halves held fast by a locking nut. The camshaft carries bevel gears imparting motion to the starting air distribution valve, speed governor and fuel booster pump. The camshaft is driven by the crankshaft by

way of an idler gear. The booster pump feeds fuel to the fuel-injection pump under a pressure of 4.9×10^4 N/m^2 (0.5 kgf/cm^2).

A facility for cleaning the fuel from sediments consists of a duplex primary filter, a duplex secondary filter and edge-type filters, each fitted with injectors. Each duplex filter consists of two sections which function independently of each other and are fitted with filtering elements made from wire gauze and felt, respectively.

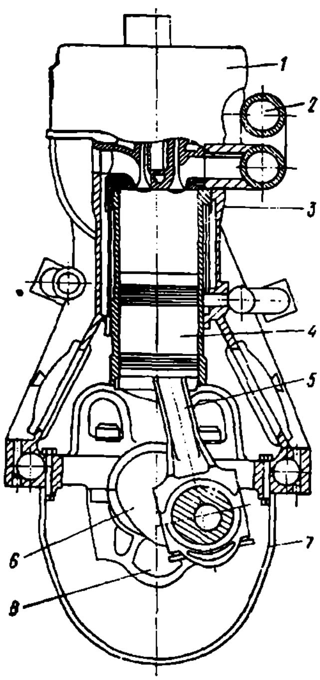


Fig. 140. Type ЧН 30/38 diesel engine

1—cover; 2—exhaust manifold; 3—cylinder liner; 4—piston; 5—connecting rod; 6—crankshaft; 7—oil sump; 8—main bearing cap

Each of the cylinders is provided with a fuel-injection pump of its own. The mode of fuel regulation is by timed by-pass. Each injector of the valve-closed type has eight orifices with a diameter of 0.35 mm. The spray angle is 140 deg, and the valve-opening pressure is $1\,960 \times 10^4$-$2\,100 \times 10^4$ N/m^2 (200-205 kgf/cm^2). The centrifugal speed governor is of the overspeed type. The scavenging pump feeding air into the receiver is of the reciprocating type with automatic inlet and delivery valves. The 6ДН 30/50 diesels are equipped with the TK-30 turbocharger delivering the air into the reciprocating scavenging pump which feeds it into the receiver under the pressure of 15.7×10^4 N/m^2 (1.6 kgf/cm^2) through the cooler.

The full-force-feed lubricating system is operated by the crankshaft-driven gear-type pump feeding oil through the tube-and-shell cooler, wire-gauze primary filters and fine filter with paper filtering elements contained in two separate sections. The cylinder liners, starting air distribution valve, scavenging pump rod and its plain bearing in the partition are lubricated with the aid of the lubricators actuated by the scavenging pump crosshead through the intermediary of levers and pivoting tie rods.

Two centrifugal pumps geared to the crankshaft are used to circulate fresh water through the engine and outboard water through the water cooler. A thermostat is provided at the outlet from the engine to control the jacket-water temperature depending on engine load. The starting is effected with compressed air at 118×10^4-294×10^4 N/m^2 (12-30 kgf/cm^2), using the control lever which also functions as a reverse control lever.

The engines are adapted for remote control over a distance of 30 m. The alarm and monitoring system keeps check on inlet and outlet jacket-water temperature, oil temperature, oil and fuel pressure in the systems.

108. Type ЧН 30/38 Diesel Engine

The ЧН 30/38 diesel (Fig. 140) is designed for use as main engine. Its leading particulars are as follows:

Number of cylinders	6 or 8
Power, kW (hp)	1 030-1 840 (1 400-2 500)
Cylinder bore, mm	300
Piston stroke, mm	380
Mean effective pressure, N/m² (kgf/cm²)	110×10^4-163×10^4 (11.20-16.65)
Specific fuel consumption, g/kW h (g/hp h)	215-218 (158-160)
Overall dimensions, mm:	
length	4 604-5 670
width	1 316
height	2 883-2 880
Mass, dry, kg	14 500-19 000

The engine has a welded cylinder block containing cast iron liners. The cylinder head is in heavy-duty cast iron and is fitted with a starting air valve, two inlet and two exhaust valves, and an injector unit per cylinder. The oil-cooled pistons are of the built-up type, having a heavy-duty cast iron head and a grey cast iron skirt. The connecting rods are I-section steel drop-forgings. The piston pins are of the floating type. The crankshaft is drop-forged from alloy steel and has nitrided journals and crankpins. The direct-reversing models are fitted with a pneumatic reversing system employing a piston-type air motor to shift the camshaft. The variable-speed governor is a pilot-actuated one.

The full-force-feed lubricating system uses the dry-crankcase principle. The cooling is by a closed-circuit system. The turbocharger is of the 2TK type.

109. Type ЧРН 36/45 and ЧРПН 36/45 Diesel Engines

The direct-injection ЧР 36/45 four-stroke finds application as main engine and is characterized by the following data:

Builder's designation	Г-60	Г-70
Number of cylinders	6	6
Power, kW (hp)	663 (900)	884 (1 200)
Crankshaft speed, rpm	375	
Cylinder bore, mm	360	
Piston stroke, mm	450	

Mean effective pressure, N/m^2 (kgf/cm^2)	77×10^4 (7.86)	102×10^4 (10.48)
Specific fuel consumption, g/kW h (g/hp h)	224	218
	165+5%	160+5%
Overall dimensions, mm:		
length	5 665	5 628
width	1 785	1 755
height	3 395	3 395
Mass, dry, kg	27 000	22 800

The crankcase/cylinder block is in cast iron, the bedplate is of a welded construction. The main bearings with steel babbitt-lined shells are supported by transverse members of the bedplate. The cylinder liners are made of cast iron and so are the cylinder heads, each superposing a cylinder. The cooled pistons in gray cast iron are provided with concave heads, each of which bears four compression rings and four oil-scraper rings. The floating piston pins are bored hollow and case-hardened. A bronze bushing is press-fitted into the small end of each I-section drop-forged connecting rod, the big end is split and contains steel babbitt-lined shells. The drop-forged crankshaft is made from alloy steel and carries balance weights.

For each cylinder there is a fuel-injection pump, and fuel regulation is accomplished by timed by-pass effected by rotating the pumps plunger with the scroll around its axis. The nozzle valves of the injectors are opened hydraulically. Sediments are removed from fuel by a wire-gauze filter and a filter with a corrugated calico filtering element. The dry-crankcase lubricating system incorporates two gear-type oil pumps, an oil cooler, primary and fine oil filters and a thermostat.

The engine is cooled by a closed-circuit system circulating through which is fresh water and this, in its turn, is cooled by the outboard water. Scavenging air and crankcase oil are also cooled by the outboard water. Both cooling systems are operated by centrifugal pumps. For starting, use is made of compressed air under a pressure of 98×10^4-118×10^4 N/m^2 (10-12 kgf/cm^2). The turbocharger is of the TK-30 type.

Engine operation can be controlled either from the local console or from the remote control system.

110. Type ДКРН Diesel Engines

Intended for the use in marine application as direct-driving main engines, the ДКРН diesels are characterized by the following data:

	5ДКРН 62/140-3	6ДКРН 74/160-3	8ДКРН 74/160-3	12ДКРН 80/160-4
Number of cylinders	5	6	8	12
Power, kW (hp)	4 500 (6 100)	7 800 (10 600)	10 100 (13 700)	21 100 (28 800)
Crankshaft speed, rpm	140	120	120	122
Cylinder bore, mm	620	740	740	800
Piston stroke, mm	1 400	1 600	1 600	1 600
Mean effective pressure, N/m^2 (kgf/cm^2)	91.2×10^4 (9.3)	94×10^4 (9.6)	91.2×10^4 (9.3)	106.2×10^4 (11.0)
Specific fuel consumption, g/kW h (g/hp h)	210 (155)	210 (155)	210 (155)	210 (155)
Overall dimensions (reverse and reduction gear included), mm:				
length	9 390	12 770	15 490	20 715
width	3 570	4 070	4 070	4 070
height	9 575	11 130	11 130	11 130
Mass, dry, kg	225 000	382 000	500 000	935 000

A feature of the engines are A-frames of the crankcase fitted to which are removable inspection doors. The bedplate is of a welded construction, its transverse beams give support to the main bearings. The welded oil pan is attached to the lower part of the bedplate. The scavenging air receiver is provided with a separate compartment containing the camshaft chain drive. The bottom of the scavenging air receiver, or entablature, forms the top of the crankcase. Glands are provided at points where the piston rods intersect the entablature. The A-frames support the cast iron cylinder jackets. The cylinder liners are in alloy cast iron. The cylinder heads are cast from steel. Each head is fitted with two injectors, an exhaust valve, an inlet valve, an indicator cock and a relief valve capable of initiating an alarm. The pistons are oil-cooled by way of telescopic pipes. Each of them consists of a steel drop-forged head and a cast iron skirt. There are five compression rings and one oil-scraper ring per each piston.

The connecting rods are steel drop forgings connecting by their jaws to the crosshead slippers lined with babbitt at both sides. The big ends are connected to the crankpins. The steel crankshaft is of built-up construction. The camshaft, chain-driven from the crankshaft, serves to actuate the exhaust valves and fuel-injection pumps.

The cylinder block, scavenging air receiver, crankcase frames and bedplate are tied together by through bolts. The pendulum speed governor is of either the overspeed or the variable-speed type. The fuel system incorporates fuel booster pumps, fine fuel filters with filtering elements in porous bronze, fuel-injection pumps, injectors with edge-type filters, a steam preheater for residual fuel with an automatic viscosity controller and a fuel cooler for cooling the in-

jectors. The fuel-injection pumps, one per each cylinder, are of the type known as pumps with plunger-controlled ports.

The lubricating system consists of a number of self-contained circuits feeding oil to the turbocharger bearings, fuel-injection pump drives, exhaust valve gear, camshaft bearings, cylinder liners and connecting-rod crank mechanism. Each of the circuits is provided with a pump to feed the oil.

The cooling system is a closed-circuit one. The pulse turbocharger is actuated by the exhaust from three or four cylinders. An air cooler is used to cool scavenging air before feeding it into the receiver. The temperature of jacket water at the outlet from the diesel and that of oil at the inlet are controlled automatically, using thermostats. An alarm and monitoring system is also provided on each engine.

111. PC-SEMT Pielstick Diesel Engines

The turbocharged medium-speed four-stroke diesels made by Pielstick of France (see Table 4) are widely used nowadays as main

Table 4

Principal Particulars of Pielstick PC-SEMT Diesel Engines

Designation	Power per cylinder, kW (hp)	Crank-shaft speed, rpm	Number of cylinders	Cylin-der bore, mm	Piston stroke, mm	Mean effective pressure, 10^4 N/m^2 (kgf/cm^2)
PC 2-2 L-400	368 (500)	500	6, 8, 9	400	460	191 (19.5)
PC 2-2 V-400	368 (500)	500	10, 12, 14, 16, 18	400	460	191 (19.5)
PC 2-3 L-400	394 (535)	500	6, 8, 9	400	460	191 (19.5)
PC 2-3 V-400	394 (535)	500	10, 12, 14, 16, 18	400	460	191 (19.5)
PC 2-5 L-400	480 (650)	520	6, 8, 9	400	460	191 (19.5)
PC 2-5 V-400	480 (650)	520	12, 14, 16, 18	400	460	191 (19.5)
PC 3 L-480	700 (950)	470	6, 8, 9	480	520	189 (19.3)
PC 3 V-480	700 (950)	470	12, 14, 16, 18	480	520	189 (19.3)
PC 4 V-570	1 100 (1 500)	400	10, 12, 14, 16, 18	570	620	209 (21.3)

engines of various ships. The key to their success is good economy, stemming from the use of residual fuel, and automatic operation.

The engines feature a robust frame of a welded construction. Steel arches are used to support the main bearings. The cylinder heads, cylinders and crankcase are tied together by through bolts. Each of the cast iron octagonal cylinder heads is fitted with two inlet and two exhaust valves, an injector and a relief valve. The cylinder head is cooled by water. The valves are actuated by the camshaft operating in conjunction with tappets, push rods and rocker arms. The exhaust valves are provided with Rotocap rotators.

The oil-cooled pistons are made from an aluminium alloy, and each is fitted with four compression rings and two oil-scraper rings which are all located above the piston pin. The oil-scraper rings are of the double-hook spring-loaded type. The cylinder liners are surrounded by water jackets. The connecting rods are I-section drop forgings. The steel one-piece crankshaft is also drop-forged. Grooves provided in webs serve to accommodate balance weights. A spring-type vibration absorber is fitted to the fore end of the crankshaft. The thin-shelled steel crankpin bearings and the leaded bronze-lined main bearings are coated with a thin layer of tin. The crankshaft-driven camshaft is located outside the engine frame and serves to actuate the inlet and exhaust valves and the fuel-injection pump. The fuel regulation is by timed by-pass. The injectors are cooled with water. The in-line engines are fitted with one turbocharger, the Vee-type ones make use of two turbochargers.

The engines operate on viscous fuel. For their reversing, the camshaft is to be shifted axially. The water pumps are of the centrifugal type, the oil pump is a two-lobe rotary one and the fuel booster pump is of the gear type.

112. MAN Diesel Engines

The diesel-building firm MAN from the Federal Republic of Germany, specializes in single-acting crosshead two-strokes of the direct-reversing turbocharged type, and in medium-speed four-strokes (Table 5).

Consider some design features of the low-speed two-stroke diesels. The welded bedplate in steel fitted with the main bearing seats gives support to the A-frames forming the engine frame. Through bolts are used to interconnect the bedplate, frame and cylinder blocks. Each cylinder block consists of separate cylinders with cast iron liners pierced with exhaust and scavenging ports, all located at one side. To prevent losses of air, the rotary valves, provided in the exhaust manifold, are arranged to close the exhaust ports until the scavenging ports are covered by the piston displacing towards TDC.

The cylinder liners of the engines with a cylinder bore over 600 mm are made up of two parts each, the topmost one is water-cooled. The cylinder heads are also of composite construction, each consisting of a lower toroidal part made of steel and an upper cast iron part designed for robustness. A hold-down ring joins the two parts.

The pistons of the MAN KZ engine feature cast iron skirts girdled with leaded bronze rings. The piston of the K6Z 57/80C diesel has two guide rings in leaded bronze press-fitted into the upper part of the skirt and one, into the lower part. The K7Z 70/120 engine is fitted with only one guide ring in the lower part of the piston skirt.

Table 5

Principal Particulars of MAN Diesel Engines

Designation	Number of cylinders	Power per cylinder		Crankshaft speed, rpm	Mean effective pressure	
		kW	hp		10^4 N/m^2	kgf/cm^2
		Low-speed engines				
52/105 B	5-9	885	1 200	183	127	13.0
70/125 B	5-10	1 520	2 060	145	128	13.1
78/155 B	6-9	1 960	2 660	122	127	13.0
90/160 B	6-10; 12	2 700	3 660	122	127	13.0
52/105 BL	5-9	885	1 200	165	141	14.4
70/125 KSZ	5-10	1 520	2 060	130	143	14.6
78/155 BL	6-9	1 960	2 660	110	141	14.4
90/160 KSZ	6-10; 12	2 700	3 660	110	142	14.5
		Medium-speed engines				
ASL 25/30	6; 8; 9	200	272	1 000	159	16.3
ASL 25/30	12; 16; 18	200	272	1 000	159	16.3
L 32/36	6; 8; 9	370	503	750	199	20.3
V 32/36	12; 14; 16; 18	370	503	750	199	20.3
L 40/45	6; 8; 9	550	750	600	191	19.5
V 40/45	12; 14; 16; 18	550	750	600	191	19.5
L 52/52	6-9	885	1 200	514	183	18.7
V 52/52	10; 12; 14; 16; 18	885	1 200	514	183	18.7
V 65/65	12; 14; 16; 18	1 325	1 800	400	181	18.43

Compression rings fitted into the piston head vary in the number between five and six depending on engine model. Oil-scraper rings are not used. The pistons with a diameter under 700 mm are cooled by oil, and those above, by water. Each piston rod connects to the guide shoe. The small end of the connecting rod round in section is of a jawless construction. Two bearing bolts connect the big end to the shank of connecting rod. The air-tight diaphragm separates the cylinder bores from the crankcase.

The crankshaft consists of two sections joined by flanges. Each crankpin is made integrally with the webs and the journals are press-fitted into the webs.

The engines employ loop scavenging. The turbocharging system varies with engine model. On most engines preference is given to the combined system of turbocharging with a pulse gas flow. Also in use are parallel and successive supercharging. For trouble-free starting and carrying fractional loads, the recourse is made to an extra electrically-driven blower.

The fuel-injection pumps feature plunger-controlled ports. The injectors are cooled with water. The camshaft serving to actuate the

fuel-injection pump and the spool valves of the starting air distributor is set into motion by a chain drive connected to the crankshaft. Hydraulic cylinders are used to shift the camshaft axially for reversing the engine.

Lubricating oil is circulated with the aid of a self-sustained electric pump. The cooling system is a closed-circuit one, fresh water being used to cool the engine, turbochargers, pistons (on the KZ 70/120C), oil and air.

113. Sulzer Bros Diesel Engines

Swiss-made Sulzer engines are widely used on board many ships as main propulsion installations and for auxiliary work. The maximum power output per cylinder of a mass-produced RD-90 diesel is 1 690 kW (2 300 hp). The design cylinder rating of a RND-105 engine is 2 940 kW (4 000 hp) at 103 rpm so that the total power of the 12-cylinder unit is 35 400 kW (48 000 hp). For the principal particulars of Sulzer diesel engines see Table 6.

Consider the construction of Sulzer engines as exemplified by the 6RSAD-76 and RD-90 models. The bedplate and cylinder columns of the engines are weldments. The cylinder block is secured to the bedplate by through bolts. A diaphragm with stuffing boxes for the piston rods completely separates the crankcase from the cylinders. The cast iron cylinder liners are provided with fins on the outside of the upper part to enhance heat transfer. They rest each on a steel spacer ring superposing the cylinder block and serving to relieve the liner of excessive mechanical and thermal stresses. The inside of the liner is protected against impinging fuel jets by the fire ring. The exhaust ports are located at the top and the scavenging ports, at the bottom of the liner.

The cylinder heads are of concentric construction intended to reduce thermal stresses. The outside toroidal part in steel and the inside insert in cast iron have different coefficients of linear expansion. The cylinder head of the RND diesel is a drop-forging. It is cooled by water and has an injector, a non-return starting air valve of the poppet type, a relief valve, and an indicator cock.

The piston of the RND engine is provided with a skirt which is somewhat longer than this is the case on the RD diesel. This permits the elimination of the rotary valves closing the exhaust ports when the piston is at TDC. The pistons are cooled with water.

On the RD engine the pistons are of the built-up type, having each a steel head and a short cast iron skirt which are both attached to an upper flange of the piston rod. The head is of a section thinner than on other engines and is reinforced with ribs in order to sustain the mechanical and thermal stresses inherent on large-bore diesel

Table 6

Principal Particulars of Sulzer Bros Diesel Engines

Designation	Power per cylinder		Crankshaft speed, rpm	Cylinder bore, mm	Piston stroke, mm	Number of cylinders	Mean effective pressure		Mass, t
	kW	hp					10^4 N/m^2	kgf/cm^2	
RD-44 DP44/76	332-368	450-500	215	440	760	5-12	80-88.8	8.15-9.06	81-167
RD-56 DP56/100	560-614	760-833	170	560	1 000	5-12	80.5-88	8.2-9.0	145-313
RD-68 DP68/125	808-880	1 100-1 200	135	680	1 250	5-12	79-86.6	8.07-8.83	208-454
RND-68M DP68/125	1 400	1 900	150	680	1 250	5-12	123	12.56	—
RD-76 DP76/155	1 100-1 180	1 500-1 600	119	760	1 550	4-12	7.9-84.3	8.07-8.6	300-680
RND-76M DP76/155	1 765	2 400	122	760	1 550	5-9	123.5	12.59	—
RD-90 DP90/155	1 690	2 300	119	900	1 550	6-12	86.2	8.82	500-940
RND-90M DP90/155	2 454	3 350	122	900	1 550	6-12	122.8	12.53	—
RND-105 DP105/180	2 940	4 000	107	1 050	1 800	8-12	105	10.7	—
V65/65	1 590	1 800	400	650	650	12-18	180	18.4	—
Z40/48	535	725	530	400	480	6-18	196	20.0	—

engines. The short skirt is conducive to a smaller overall height of the engine. The piston head is fitted with six compression rings having bevelled joints. The piston cooling is effected with the water fed over telescopic pipes. On completing its task, the water is disposed of through a pipe and a pivotal arrangement of the crosshead.

The pistons of the RD-76 and RD-90 engines are water-cooled, for the cooling with oil creates the hazard of a rapid carboning up of the piston head from the inside. To prevent an ingress of water into the crankcase, the telescopic pipes feeding water to the piston are contained in a water-tight sleeve. The engine cooling is taken care of by a closed-circuit system.

The RD diesel engines feature built-up crankshafts. Each crankpin is made integrally with the webs and the journals are press-fitted into the webs. The crossheads are of the rectangular cross section and with only one shoe (on the RD-76 and RD-90 engines, each crosshead is fitted with two shoes); they link the piston rods to the steel connecting rods with a massive flange at the end attached to which are two crankpin bearings. The reciprocating duplex fuel-injection pumps employ fuel regulation by changing the points of the beginning and the end of delivery, or by timed by-pass. They are actuated by symmetrical cams used for running ahead and astern.

The valve-closed injectors have each ten orifices with a diameter of 0.9 mm and afford an injection pressure of 2 640 N/m^2 (270 kgf/cm^2). The speed governors are of the variable-speed type.

The camshaft is chain-driven. For reversing the engine, the camshaft and the spool valves of the starting air distribution valve are to be shifted axially. On the RD-76 and RD-90 engines, a gear train is substituted for the chain drive. In addition to being shifted axially preparatory to a reversing maneuver, the camshaft is also turned through a certain angle with respect to the crankshaft (on the RD-90 diesel, this angle is 98 deg).

The engines are provided with alarm and monitoring systems shutting down the engine in case of low jacket-water pressure or low oil pressure. The scavenging is of the cross loop pattern with controllable valves in the exhaust duct of each cylinder. The valves, which are either of the rotary or pivoted type, serve to time the exhaust (pivoted valves are used on the RD-90 engine). The scavenging system is a combined, two-stage facility with centrifugal blowers functioning as the first stage, and the displacement effect of the piston underside space as the second stage. The scavenging air pressure is 15.7-17.6 N/m^2 (1.6-1.8 kgf/cm^2). The number of turbochargers used on each engine depends on the number of cylinders, the 8 RSAD-76 diesel having thus two and the 9RD-90 engine, three setups.

114. Burmeister and Wain Diesel Engines

There is a great many of ships using direct-coupled Burmeister & Wain engines as the main propulsion plant. For the principal particulars of these Danish-built diesels see Table 7.

Table 7

Principal Particulars of Burmeister & Wain Diesel Engines

Designation	Power per cylinder		Number of cylinders	Crankshaft speed, rpm	Mean effective pressure	
	kW	hp			10 N/m²	kgf/cm²
42VT2BF-90 (ДКРН 42/90)	368	500	5-12	210	84.3	8.6
50VT2BF-110 (ДКРН 50/110)	515	700	5-12	170	84.3	8.6
62VT2BF-140 (ДКРН 62/140)	800	1 090	5-12	135	84.6	8.6
74VT2BF-160 (ДКРН 74/160)	1 100	1 500	5-12	115	84.3	8.6
84VT2BF-180 (ДКРН 84/180)	1 550	2 100	5-12	110	84.3	8.6
98VT3BF-200 (ДКРН 98/200)	2 430	3 300	6-12	100	96.5	9.85

Consider by way of an example, the design features of the 74VTBF-160 engine. It is a low-speed two-stroke of the crosshead type with a uniflow valve and port scavenging. The engine is a direct-reversing supercharged one. The engine structure comprises a welded bedplate, A-frames and cylinders which are tied together by through bolts passing via sleeves welded into the A-frames. The cylinder liners are provided with sealing collars at their tops which support them in the water jackets. The lower ends of the liners extend into the air receiver and are pierced with scavenging ports spaced equidistantly around the entire circumference.

Each cylinder is overlaid with a dome-shaped cylinder head in steel which is fitted with an inlet valve and an exhaust valve, a relief valve, an indicator cock and injectors. The water-cooled exhaust valve is actuated by a symmetrical cam of the camshaft. The exhaust valve discs are hard-faced for wear resistance, and a clearance absorber is incorporated into the valve gear which allows the valve stem to expand with an increase in temperature. Diaphragms with stuffing boxes for the piston rods separate the crankcase from the cylinders. The double bottom of the scavenging air receiver overlaying the crankcase protects this against any increase in temperature which may occur if the oil accumulated in the receiver catches fire. On earlier models of the engine, the heating-up of the bottom created the hazard of an oil mist explosion.

The steel crankshaft features built-up crankpins joined with the aid of flanges. The steel shells of the main and crankpin bearings are lined with babbitt. The built-up pistons cooled with oil consist of

a steel head and a cast iron skirt each. Six compression rings are fitted into the piston head. The cooling oil is fed to the crosshead over a telescopic pipe, reaching then the piston head by way of the piston rod. It is drained through the clearance around the delivery pipe.

The steel connecting rods are of a round section. Scavenging air is fed by two pulse-type turbochargers, each driven by the exhaust from three or four cylinders. A plate heat exchanger is used to cool the air. Each of the cylinders is served by a high-pressure fuel-injection pump with plunger-controlled ports delivering to two cooled injectors having each three orifices with a diameter of 1.1 mm. The injection pressure is 2 450 $\times$ 10^4 N/m^2 (250 kgf/cm^2). A gear-type fuel booster pump feeds fuel into the fuel-injection pumps through a felt filter. A preheater is provided before the fuel-injection pumps if the engine uses residual fuel. Centrifugal separators and settling tanks are also included into the fuel system in this case. The camshaft actuating the fuel-injection pumps is set into motion by the crankshaft through a twin-strand chain drive. On later models as, for example, the 84VT2BF-180 one, the fuel-injection pumps and exhaust valves are set into motion by the same camshaft.

The engine cylinders, exhaust valves, cylinder heads and turbochargers are cooled by a closed-circuit system. The pistons are oil-cooled. For speed governing use is made of a variable-speed governor. An individually-driven pump induces the circulation of oil in the lubricating system. The cylinder liners are lubricated by means of lubricators. The starting is by compressed air. The engine controls consist of a fuel-regulating handle and an ahead and astern handle. For reversing the engine, its speed is reduced to 60-90 rpm. At 40 rpm, air is admitted into the engine cylinders which brings them at rest in 2-3 s. A 10-percent engine overload is tolerable without a time limit, a 20-percent overload can be carried not over 1 hour. The most recent Burmeister & Wain's engines of the LGF range display a cylinder rating between 650 and 3 000 kW.

This country has obtained the license from Burmeister & Wain to manufacture diesel engines of the 7ДКРН 74/160 and 5ДКРН 50/110 type with an increased supercharging level of 16.2 N/m^2 (1.65 kgf/cm^2) instead of 13.2 N/m^2 (1.35 kgf/cm^2). The mean effective pressure has been increased from 69.8 $\times$ 10^4 N/m^2 (7.12 kgf/cm^2) to 84.3 $\times$ 10^4 N/m^2 (8.6 kgf/cm^2). On the 9 ДКРН 80/160 diesel, the mean effective pressure will be 103 $\times$ 10^4 N/m^2 (11.0 kgf/cm^2).

REVIEW QUESTIONS

1. What cooling systems are employed on the diesel engines discussed in this chapter?
2. What types of bearings are used predominantly on the diesel engines discussed?
3. Calculate the power of the diesel engines referred to in the tables using the data given there.

Chapter XX

PROPULSORS

115. Transmission of Power From Engine to Propulsor

The ship's shafting serves to transmit the rotative power from the main propulsion engine to the propulsor and also the thrust developed by the propulsor to the ship's hull. The shafting consists of several shafts rigidly attached to each other and supported in bearings resting on special supports termed foundations.

As it can be seen from Fig. 141, the shafting comprises thrust shaft *3*, intermediate shaft *2* and propeller shaft *1* a section

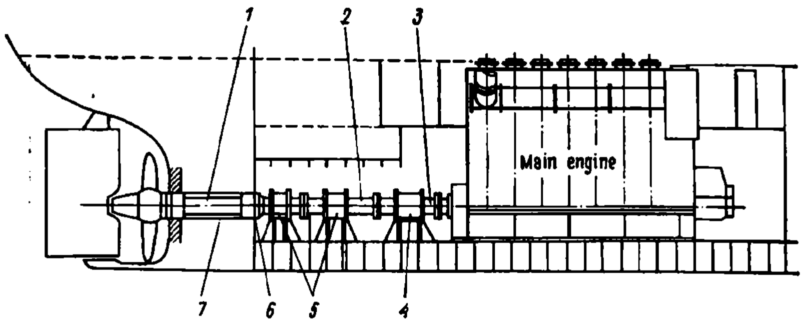


Fig. 141. Shafting arrangement typical of single-shaft ships

whereof passing through the stern tube is referred to as the stern tube shaft. The intermediate shaft is supported in line-shaft bearings *5* and the propeller shaft, by stern tube *7*. Serving to seal off the place where the propeller shaft leaves the ship's hull, the stern tube is one of the most crucial components of the shafting. The stern-tube bearing *6* has inserts commonly made from lignum vitae (a kind of hardwood). Also in use are rubber compound bearings and babbitt-lined bearings. They are lubricated with the outboard water. The packing gland, the stern tube is fitted with, consists of braided grease-impregnated hemp.

Some modern ships employ Simplex stern-tube packing arrangements consisting of two glands applied to the fore and aft end of the stern tube. The ingeniously-shaped glands in a rubber compound are lubricated with liquid mineral oil. In this case there is an oil tank for lubricating the propeller shaft passing through the stern tube and another tank for feeding oil to the fore gland. Stern-tube

shafts need no bronze cladding and work in babbitt-lined sleeves lubricated with mineral oil. The aft gland works in water and is completely water-tight.

Main thrust bearing *4* gives not only an axial support to the thrust shaft but also transmits the propeller thrust to the hull.

Designed to take axial loads, the thrust bearings in use feature one or two bearing rings. A single-ring thrust bearing is fitted with tiltable babbitt-lined shoes on either side of the ring which function in running ahead and astern. A resultant of the thrust taken up by each shoe when the propeller shaft is in operation causes the shoe to tilt to that a wedge-shaped oil film capable of supporting unit loads as high as 245×10^4-346×10^4 N/m^2 (25-35 kgf/cm^2) is automatically formed between the shoe and shaft collar.

In modern motor ships the main engine may be linked to the propeller shaft either directly or indirectly. In the former case the engine shaft connects to the propeller shaft by means of a rigid or flexible coupling only. In the latter case, the recourse is to a mechanical arrangement or an electric drive. The mechanical arrangement takes the form of speed-reducing gears transmitting the engine power to the propeller shaft which rotates at a speed lower than that of the engine shaft. Gear ratios vary in marine practice between 2 : 1 and 4.5 : 1.

116. Screw Propellers: Elements and Design Features

The contrivance which utilizes the work performed by the main engine in order to produce the thrust capable of propelling the ship in a given direction is termed propulsor. Depending on the design, propulsors fall into screw propellers and water jets. The former produce thrust by driving water in the direction opposite to the direction of ship's advance and enjoy wide-spread application. The latter produce thrust by discharging water at a high velocity with the aid of a pump and are used in smaller craft.

Any screw propeller may be characterized in terms of its diameter D, pitch H, the number of blades, pitch ratio and efficiency (Fig. 142).

The diameter of a propeller is that of a circle described by the tip of a propeller blade. The pitch is the distance through which the propeller advances in the course of one complete revolution in an unyielding medium. For a dimensionless expression of the pitch, P, use is made of the pitch ratio, P/D. Its value varies depending on the type of work the ship is intended to cope with. In high-speed applications the pitch ratio exceeds 1.5, but smaller values are practical for towing or trawling operations in order to obtain the highest possible propeller efficiency. This, in its turn, is the ratio of the power absorbed to produce the thrust and the power delivered to the propeller.

The overall efficiency of a diesel engine, propeller and ship's hull performing together is expressed in terms of propulsive efficiency (also called propulsive coefficient) which is the ratio of the effective power required in towing the hull to the indicated or shaft power delivered to the propeller. The propulsive efficiency varies between 0.35 and 0.8, increasing directly with an increase in the efficiency obtained in utilizing the power delivered to the propeller by the engine.

Ships make use of propellers of two different types which are constant-pitch (CP) propellers and controllable-reversible-pitch (CRP)

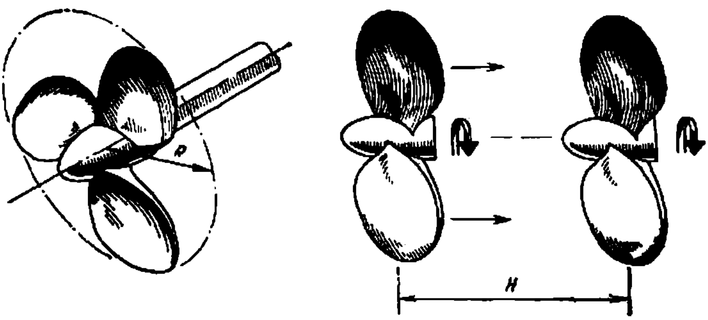


Fig. 142. Elements of screw propeller

propellers. In the former, the pitch of the blades at all their sections is always the same. In the latter, the pitch may vary both with the radius and section of a blade.

CP propellers cannot assure good maneuverability of the main engine and complete utilization of its output under any load and speed conditions; they also create problems in taking off engine power to set into motion auxiliaries. Meanwhile, a maximum propulsive efficiency is obtained when the characteristic curve of the engine and that of the propeller coincide. To that end, which enables a complete utilization of the engine power output without the risk of overloads, ships employ CRP propellers.

With these propellers one can easily meet any increase in the resistance to ship motion through the water, which is likely to be encountered in trawling or towing, or due to the fouling of the hull or bad weather, by changing the propeller pitch so that the two characteristic curves coincide. CRP propellers are most suitable for fishing craft, surveying vessels and other ships whose operation involves frequent reversing which is effected in this case by turning the propeller blades. Since the ship need not be stopped, the maneuver is rapid and the engine is protected against extra wear. Also

the period of setting a ship underway is materially reduced and so is the period of stopping her, obviously because a CRP propeller can shorten the stopway by 25%.

A CRP propeller is a screw propeller in which the blades are separately mounted on the hub, each on an axis, and in which the pitch of the blades can be changed by means of an internal mechanism, and even reversed, while the propeller is turning. However, CRP propellers are not free from some drawbacks which limit the field of their application such as complicated construction—particularly this applies to the pitch-changing mechanism with mechanical, hydraulical and electrical elements—high cost and the need in skilled attendance.

Screw propellers are made from bronze, brass, non-ferrous steel, cast iron and plastics. In applications where high propeller loads are involved, as this is the case with trawlers and tugs, the screw propeller may operate in a metal ring or nozzle attached to the hull at the top. The longitudinal sections are of airfoil shape, its curved side constituting the inside surface of the nozzle. The cross-sectional area of the nozzle entrance is made from 50 to 60% bigger than that of the nozzle exit so that more water is drawn to augment the thrust, as compared to the open propeller. In some highly-maneuverable ships, the whole nozzle is made pivotable, becoming thus a very efficient steering mechanism capable of changing the direction of the water flow produced by the propeller.

117. Performance of Main Engine and Propulsor

The main propulsion plant, which may be either directly-coupled or geared to the screw propeller, is a constituent part of a hydrodynamic complex comprising the engine, ship's hull and propeller whose elements are all interconnected.

The power absorbed by the screw propeller varies with the ship's speed and draught, the resistance to ship motion through the water, the state of sea, the direction and force of wind and other particulars of the sailing conditions such as the presence of ice, traversing shallows, etc. Particularly unfavourable are regarded to be those operating conditions which involve reversing, the steaming at a minimum stable shaft revolutions, towing operations and the navigation in rough seas. To evaluate the efficiency achieved in utilizing a ship, it is necessary to take into account the practice of operating her engine with an eye to its good economy along with durability and reliability. These factors are significantly influenced by the mode of running the engine as prescribed by its characteristic curves establishing a functional dependence of engine parameters upon a quantity adopted as an independent variable. So, a characteristic speed

curve establishes the relationship between the main engine parameters and the rate of crankshaft rotation.

Distinction is made between external characteristic speed curves and propeller characteristic speed curves. An external characteristic curve represents the main engine parameters as a function of the rate of rotation with the fuel regulating lever being set into a certain position for carrying the load. This curve may be referred to the gross power, overload power, or rated power. When an engine operates at a maximum fuel delivery corresponding to the highest possible mean effective pressure, the external characteristic curve depicting the mode of operation is referred to as a gross one. This operating extremity is prohibited for use, causing overheating of the cylinders and pistons, excessive fuel consumption and smoky exhaust.

The operation of an engine with the fuel regulating lever being set so as to enable the engine to develop short-time overload power is depicted by an overload external characteristic curve. The operation of an engine without any time limit under the conditions permitting the main engine parameters to be maintained as prescribed by the data entered into the engine certificate is represented by an external characteristic curve for rated power. A point to be noted is that not infrequent are the cases when the service power recommended by the builder for use is somewhat lower than the rated output.

Graphically, an external characteristic curve is an almost straight line, its shape and slope being decided by the type of engine (the higher the speed of the engine, the steeper the curve).

As already mentioned, the performance of a marine engine is evaluated on the basis of a characteristic curve representing the performance of the engine and propeller together. A curve depicting engine power as a function of the rate of rotation obtained in actuating the propeller is termed propeller characteristic curve.

The power absorbed by the propeller increases directly with the rate of rotation raised to the third power

$$N_e = cn^3$$

where N_e is the power, kW (hp); c is the factor of proportionality; n is the rate of rotation, rpm.

Thus, the propeller characteristic curve is a cubic parabola.

Comparing two external characteristic curves *1* and *2* (Fig. 143) with propeller characteristic curve *3* we come to the conclusion that an engine which is being run in accordance with the propeller characteristic curve fails to operate to its full capacity while carrying fractional loads. At the same time, point A where the three curves intersect indicates a condition when the power output of the engine operated in accordance with the external characteristic curve is equal to the output obtained in following the propeller characteristic

curve. An increase in the rate of rotation in excess of the rated value causes the propeller curve to rise above the external one, indicating that the power absorbed by the propeller exceeds the engine rating. This results in an overload.

Comparing the two external characteristic curves *1* and *2* representing a slow-speed and a high-speed engine, respectively, we can see that curve *1* is less steep than curve *2*, indicating good adaptability of the slow-speed engines to varying operating conditions. It is also evident that the high-speed engines are more susceptible to changes in the rate of rotation, being underloaded at slow speeds and overloaded at high ones. This accounts for the fact that slow-speed engines are used on a larger scale in marine application than high-speed ones in spite of their obvious bulkiness.

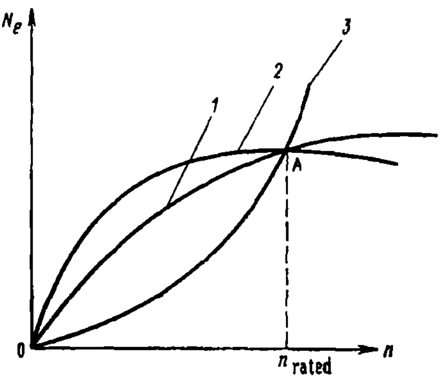


Fig. 143. Characteristic curves of main engine and propulsor combined operation

In the case of an electric propulsion drive or a diesel-generator set, the engine is operated as advised by a load characteristic curve establishing the way the main engine parameters are being influenced by the load at a constant rate of rotation. From a load curve one can find out which mode of operating the engine is the most economical one. So, it has been established that the fuel consumption is at a minimum when the engine is being run at 75-80% of its rated power output.

REVIEW QUESTIONS

1. What purpose does the ship's shafting serve?
2. What elements characterize a screw propeller?
3. How are propulsors classed with reference to the design?
4. What is the diameter of a propeller?
5. What is the pitch of a propeller?
6. What is the pitch ratio of a propeller?
7. How many blades can have a propeller?
8. What is termed the efficiency of a propeller?
9. What is the propulsive efficiency of a propeller?
10. What are the main assets and disadvantages of CP and CRP propellers?
11. What purpose do the characteristic curves of an engine serve?
12. Discuss load characteristic curves, external characteristic curves and propeller characteristic curves.

Chapter XXI

MARINE DIESEL ENGINE REPAIRS

118. Fundamentals of Repair Practice

The power output, and the rate of rotation, of a diesel engine which are the main criteria of its performance, commonly decrease after a period in service due to wear and deteriorated fits. The repair aimed at eliminating the defects consists of restoring either the fits or original dimensions of components.

The concept of planned repair calls not only for the exact scheduling of work in prospect, but also for the formulation of all the requirements for the resources. Therefore, it is essential to ascertain the scope of forthcoming repair work as accurately as possible. To that end, it is necessary to prepare the so-called repair request forms along with other appropriate documents, such as a technological map of repair operations for each engine assembly; a repair flow sheet and schedule; instructions for the dismantling, checking for defects, reconditioning, reassembling and adjusting of various engine assemblies and the engine as a whole; specifications for the manufacture of crucial engine components; description and drawings of jigs and fixtures to be used in repair; a bill of requisite materials; specifications for the running-in and testing of the engine.

Special drawings are prepared for the components which need reconditioning or must be made anew, outlining the requirements to be met in reclaiming the part, repair sizes and the allowances to be observed in fitting the new part into its place.

Preparatory to beginning a repair, the engine must be drained of oil, water and fuel. It must also be cleaned from the accumulated sludge and carbon, and stripped from the automatic control equipment and instruments.

119. Techniques of Measuring the Wear on Cylinders and Pistons

Wear is one of the factors deciding whether or not a cylinder liner is to be rejected. The most rapid wear occurs at the liner top. A worn liner bore is of elliptical configuration, for the wear in the plane through the crankshaft axis and that in the plane of connecting rod oscillations is anything but uniform, particularly on trunk engines. The two-strokes also show rapid wear next the ports of a liner which results from inadequate lubrication.

Numerous tests have proved that one starting operation gives rise to the same wear as running the engine from 3 to 5 hours.

Liner wear measurements are taken by means of inside vernier calipers and dial gauges at the points of the liner bore indicated by a template which is a steel ring with perforations and a flange supporting the template inside the liner. The perforations are arranged in the plane through the crankshaft axis and through that of connecting rod oscillations. The wear on a liner of a two-stroke diesel may be measured, e.g. in the localities along liner height as follows:

— level with the topmost compression ring while the piston being set at TDC.

— in the region of a maximum liner wear resulting from normal service;

— opposite a locating collar which may deform under thermal stresses;

— in the middle of the lower row of scavenging ports.

Piston wear is measured at the skirt, using a micrometer caliper. The measurements are taken at diametrically-opposite points at the top, middle and bottom of a plane passing through the piston pin axis, and a plane perpendicular to this axis. An out-of-round of the skirt, showing the wear due to the action of a normal component of the gas pressure, and a skirt taper both exceeding an allowable value call for scrapping the piston. The piston head commonly shows no wear, being of a diameter which is smaller than the cylinder bore.

Piston groove wear is ascertained with the aid of gauge blocks inserted into the side clearance of each groove.

An excessive wear on the outside surface of piston rings interfering with normal functioning of a piston is betrayed by an increase in the gap between the ring ends. For measuring the gap, a piston ring is to be inserted into a sleeve of an inside diameter equalling the cylinder bore, and the gap width determined, using a feeler gauge. The ring is fit for further service if the gap is within the limits specified by the builder. Each piston ring is also checked for a tight fit with the liner walls, using the same sleeve and an electric lamp. The light of the lamp must be either not seen at all between the ring and sleeve or through a clearance not wider than 0.05 mm extending not farther than an arc of 45 deg. Piston rings' distortion is checked on a surface plate, using a feeler gauge. Piston rings of a diameter between 200 and 400 mm which are in good repair will not allow a 0.05-mm feeler gauge to pass.

120. Testing Engine Components for Defects

The way any repair should be conducted is stipulated in a document called repair specifications. They cover such aspects as the description of each engine component, the procedure and techniques for defects detection, permissible dimensions of parts for further service, intolerable defects, recommended reclamation techniques.

The defects detection procedure follows the disassembling of an engine, and as an outcome of the detection the parts appear to be classified into three groups, i.e. parts fit for further service, parts requiring repair, and parts rejected to scrapping.

The defects identified in the course of detection, primarily the external and internal cracks and deformations, are entered into the repair request forms along with the exact location and measurements of each defect, and other pertinent information. The results and cost of repair are materially influenced by the scope of the data presented in these forms.

Consider some of the techniques employed in the detection of defects.

Visible flaws, such as external cracks, pitting, temper colours and the like, can be detected by visual inspection with the naked eye or using a magnifying glass. Hidden flaws are identified by means of non-destructive testing.

The most simple way of testing parts for cracks consists in that the parts are soaked in a kerosene bath for 2-3 h, wiped dry and coated with a thin aqueous chalk paste. Yellow spots appearing on the chalked surface after this has dried up will betray the cracks. This method, known as the liquid-penetrant test, is commonly used to explore babbitt-lined bearings.

Fluorescent flaw detection is instrumental in pinpointing invisible surface defects. For operation, it relies on the property of some organic compounds to fluoresce under the action of ultraviolet radiation, e.g. a mixture of coryol and kerosene emitting yellow-green light. For a test, the part is washed, degreased and coated with the solution. In 15 to 20 min, the part is again washed, dried in hot air and strewed with silica gel powder which is a drying agent. On removing the silica gel, the part is exposed to ultra-violet radiation causing the cracks, if any, to fluoresce in white.

A magnetic-particle testing consists in that a steel part is magnetized, covered with magnetic particles and then the part is flushed over with a mixture of 50-60 g of red ocher per litre of diesel fuel. The magnetic particles display a preference for adhering only at locations where the magnetic field leaks out to the surface and attracts the particles, which indicate the locations of discontinuities. After the inspection, the parts must be demagnetized.

In the case of crucial engine components, their internal defects (porosity, lack of fusion, cracks) are revealed by means of X-rays.

121. Reconditioning Engine Frame, Cylinders and Pistons

Cracks, dents and bulges in the surfaces, upon which cylinder liners rest, are typical defects of an engine frame after a period in service. Wear on the cylinder block below the top flanges of liners and cor-

rosive destruction of the surfaces in contact with the cooling water are other possible defects.

Cracks in the engine frame may vary in length and depth. They are eliminated after their causes have been analyzed, using as a rule electric arc or gas welding. Helpful in coping with the job are screw stoppers, patches, straps and clamps which are used depending on the configuration of the frame. After welding up a crack, the weld must be checked for tightness. This is done by means of a liquid-penetrant test or a hydrostatic test if the cooling system has been reconditioned.

Widely used nowadays is a speedy method of reclaiming engine frames with the aid of epoxy resin. A crack, once it has been bevelled and trimmed, is filled with a paste made up of epoxy resin and metal powder. Displaying good affinity for the metal of the frame, the paste reliably patches up the defect after curing. Sometimes a reinforcement in the form of wire or wire gauze is required in building up the cracks by means of epoxy resin.

Dents and scratches are eliminated with a scraper or a grinding wheel, and the traces of wear under liner flanges are taken care of by turning in a lathe. The surfaces in contact with the cooling water are protected against corrosion by means of anodizing.

The cause of main concern during repair operations is the condition of pistons, compression and oil-scraper rings and cylinder liners. Typical piston defects include cracked and burnt-through heads, worn skirts, corrosion. Built-up pistons having a steel head and a cast iron skirt are reconditioned by the replacement of the damaged part. A one-piece piston with a cracked head must be replaced by a new one. Scuffed pistons and liners are also subject to replacement, preferably as a pair. Minor scratching and scoring of a piston skirt and liner may be eliminated by trimming and polishing so as to give no rise to an excessive clearance between the two parts.

A wear on piston ring grooves is taken care of by turning the grooves in a lathe and fitting them with rings of the repair size.

Subject to renewal are also the piston rings in which the gap between the ends has widened 1.5 to 2 times. As far as the oil-scraper rings are concerned, renewed are those of them whose bevelled face has doubled in width.

The allowable amount of wear on the top of a liner varies with the engine speed, equalling $D/150$ for the engines with a speed under 250 rpm and $D/500$ for those developing over 500 rpm, where D is the cylinder bore. The maximum out-of-round of a liner is $D/100$ irrespectively of the speed. Heavily worn liners of big-bore engines can be reconditioned by turning and grinding.

Defective holes in piston bosses which may grow elliptical or become distorted in some other way are reconditioned by boring followed by grinding in a lathe or hand-scraping.

122. Reconditioning Fuel Equipment

The fuel equipment settings used in commissioning an engine are to be maintained throughout its period in service. To that end, they must be checked at the regular intervals as indicated in the engine service manual.

The most typical fuel system defects were discussed in Section 58.

A plunger of a fuel-injection pump which has stuck in the barrel due to being scuffed is a defect calling for replacing the pump plunger and barrel as a pair by new components. Slightly worn plungers and barrels can be reconditioned with the aid of laps, sorted by sizes in 2 μm increments and lapped to each other. Pitted and scored valves of the fuel-injection pump are turned in a lathe and ground to their seats.

The reconditioning of scuffed nozzle valves of the fuel-injection pump invites difficulties on board a ship, if is possible at all, for a stuck nozzle cannot always be withdrawn from the injector. Therefore, the injector with the stuck nozzle valve is to be renewed. In case of a sluggish nozzle valve action, the remedy is to wash the nozzle valve in diesel fuel and lap it into the nozzle body, using oil or a fine polishing paste. Plugged nozzle body orifices are cleaned with a steel needle and washed in fuel.

The after-dripping of an injector may be eliminated by lapping the nozzle valve to the nozzle body. An increase in the diameter of orifices by 10% calls for scrapping the nozzle body and so does its corrosion.

REVIEW QUESTIONS

1. What documents serve as the basis for carrying out repairs of a shipboard engine?
2. What techniques are used to determine the wear of main diesel engine components?
3. Why must the engine components be tested for defects?
4. What technique is used to recondition an engine frame?
5. What are the main defects of cylinders and pistons?
6. What defects are typical for fuel equipment? What steps are taken to eliminate them?

Chapter XXII

MARINE STEAM GENERATORS

123. Purpose and Construction of Marine Steam Generators

A steam generator is designed to produce steam under a pressure above atmospheric. From the standpoint of design, they fall into two classes: the fire-tube type and the water-tube type. In the former, the hot furnace gases pass through tubes in the water space (Fig. 144*a*), and in the latter, the water flows through tubes heated from the outside by the exhaust gases (Fig. 144*b*). Since the tubes

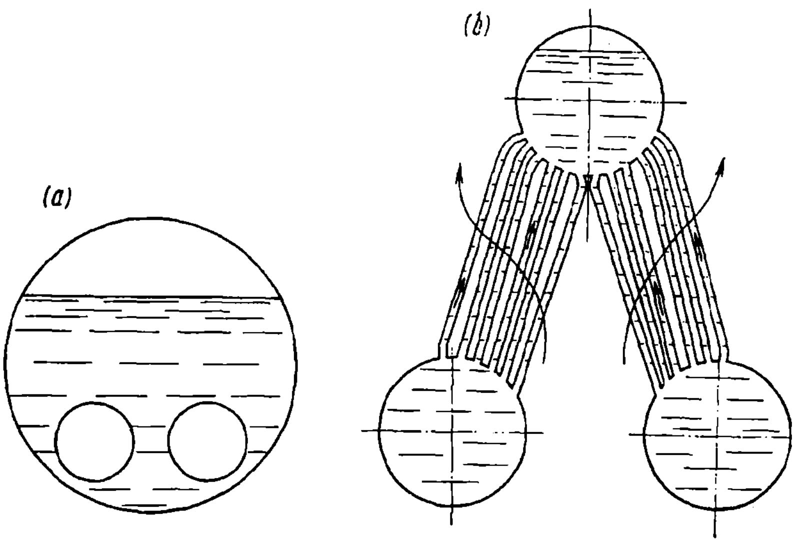


Fig. 144. Marine steam generators: (*a*) fire-tube type; (*b*) water-tube type

located close to the furnace absorb maximum heat, they generate a steam-water mixture, whereas in the distant tubes the water is below the boiling point. Thus, in operation, the system comprising the heated steam-generating tubes and the less-heated downcomers interconnected by a water drum, or header, and a steam drum forms a circulation circuit. The circulation is produced there by the difference in the densities of the water and the steam-water mixture.

In fire-tube boilers, the steam pressure is restricted to 147×10^4-166×10^4 N/m^2 (15-17 kgf/cm^2), the water-tube type is capable of

generating high-pressure steam and is therefore more efficient. Although the early marine steam generators were all of the fire-tube type only; nowadays they have been ousted by the water-tube installations which have a larger heat-generating surface and, consequently, a higher steam-generating capacity. They are less costly, display a lower mass for a given size and can be set into operation at a shorter notice than their predecessors. The steam pressure may vary in them between 15×10^4 N/m^2 (1.5 kgf/cm^2) and 988×10^4 N/m^2 (100 kgf/cm^2) or even more.

Depending on the end-use, distinction is made between main steam generators supplying steam to the main propulsion plant, and auxiliary boilers meeting the needs of marine auxiliaries and shipboard systems (sanitary, fuel viscosity control, etc.). Marine practice also knows waste-heat steam generators recovering the heat of exhaust gases. Most commonly, they are of the water-tube assisted-circulation type in which water is being heated while circulating through coiled tubes, or tube banks, heated by the gases.

In a water-tube steam generator, the water is in continuous motion. Absorbing heat from the tube walls, it warms up to the boiling point when steam bubbles begin to form. The water-steam mixture thus formed rises into the steam drum wherein also feed water is admitted at a rate keeping the water level there at mid-height. The other half is filled with steam. Since the feed water is colder than the steam, it flows into the lower drums via the tubes heated by convection, or via unheated downcommers. Accumulating in the steam drum the steam contains much moisture. To separate it, each steam generator is provided with appropriate means in the form of superheaters.

124. Fittings and Automatic Control of Steam Generators

Coming under the category of fittings are various instruments and valves provided on a steam generator to assure its safe and trouble-free operation. The fittings must meet appropriate requirements as called for in the USSR Shipping Register.

In point of application, they are classified into fittings of the steam space and those of the water space. The former consist of pressure gauges and stop valves. The latter include water gauges, plugs for drawing water for test, try cocks, surface and bottom blow valves, feed valves and drain valves.

The stop valve serves to disconnect the steam generator from the piping leading to the consumers. It is fitted at the top of the steam space and as far from the water level as possible. Steam is removed from the drum through stop valve *1* (Fig. 145) which is connected to dry pipe *2* with slots *3* permitting the steam to pass while holding back the water.

The feed valve is designed to supply water into the stem generator at a controlled rate with provision for automatic closing if the feed water pressure falls below the steam pressure. This prevents leaks of feed water through the delivery line.

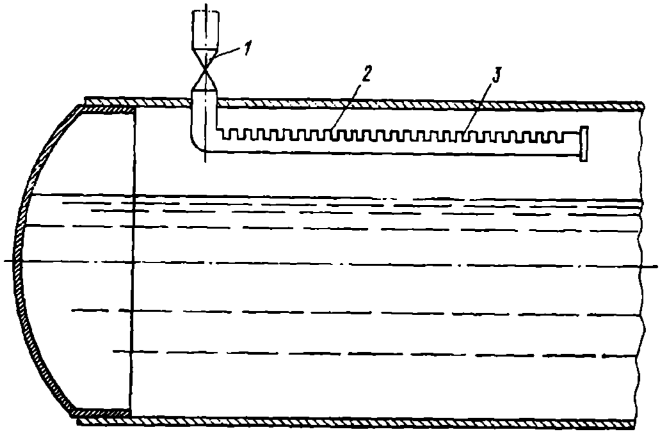


Fig. 145. Dry pipe

Water gauge *1* (Fig. 146), relying for operation on the principle of communicating vessels, is arranged to indicate the level of water in the steam generator. Doubling its function are try cocks *2* and *3*, one of them being fitted at the lowermost level of the water in the drum *4* and the other at some height above this level. Thus, it is possible to check the water level should the water gauge become inoperative.

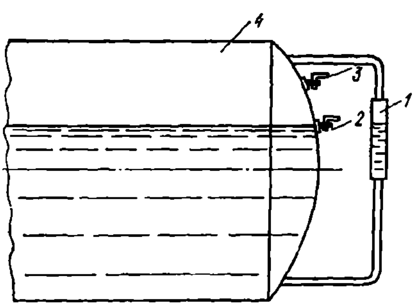


Fig. 146. Water gauge and blow valves of steam generator

The surface blow valve is intended to dispose of any grease or other impurities which may be floating on the surface of water. The purpose of the bottom blow valve is to "blow down" the steam generator and remove scale or other solids which may enter the drum with the feed water.

To prevent a buildup of an excessive steam pressure, appropriate rules stipulate that each steam generator should be provided with two spring-loaded relief valves adjustable for the opening pressure. They are set to open in response to an increase in the steam pressure by 4-5% of the allowable maximum.

A bleed valve is fitted at the top of the steam drum to enable air to escape during the start-up of the steam generator.

To enhance the efficiency of steam generator, reduce the boiler room complement and improve labour conditions, the resort is to automatic control. It maintains steam pressure and temperature at a given level, feeds air in an amount sufficient for complete combustion, keeps the water level within given limits, providing thus for optimum conditions to carry the load. The deviation of the variables from the set points is a minimum under automatic control.

Auxiliary marine steam generators are lighted off automatically after the furnace has been ventilated, and are shut down manually or under remote control. The operation of steam generators is taken care of by electrical, electromechanical and electrohydraulic systems. For example, an electromechanical system is capable of maintaining the water level and steam pressure accurately to within 50-60 mm and 35.2×10^4-44×10^4 N/m^2 (4-5 kgf/cm^2), respectively.

Modern marine auxiliary steam generators are adapted for unattended operation, and are therefore equipped with reliable alarm and protection facilities. Varying in design, the facilities are always arranged so as to cut off the delivery of fuel to the injectors. Waste heat steam generators also operate automatically, and are provided with contrivances to monitor the generation of steam and water level.

125. Some Features of Operation

Since high-pressure steam is highly hazardous for personnel, appropriate authorities (e.g. the USSR Register Inspectorate) vigilantly supervise not only the manufacture of steam generators, but also their operation and repair.

Relevant safety rules for steam and water boilers stipulate that a steam generator producing superheated steam at a temperature over 673 K (400°C) must be provided with an automatic controller of the superheated steam temperature. The water level must never be distant less than 150 mm from the highest point of the heat-absorbing surface. A steam generator with a chamber-type furnace may be operated without permanent attendance provided it is fitted with an automatic control system catering for operational supervision from a control panel, and automatic shutting down in the event of malfunction, in which case alarm signals are to be lighted at the control panel. However, the steam generator should be shut down from the control panel at any time. The events calling for a shut-down are as follows:

— over 50% of the relief valves are inoperative;

— steam pressure exceeds the permissible limit by 10%, and

tends to increase further in spite of the discontinued feeding of fuel, throttled draught, and an increased flow rate of feed water;

— the water level rapidly declines in spite of intensive replenishing;

— all water gauges are inoperative;

— cracks have been detected in crucial parts of the steam generator, breaks have been found in two adjacent strength members, etc.

To assure normal operation, each steam generator is subjected to inspections preparatory to a start-up, at regular intervals during the period in service and, if necessary, ahead of the schedule specified in the certificate. The regular inspections fall into the external inspections conducted without removing the unit from service, and internal inspections performed during outages after an appropriate preparation of the plant for the inspection. The purpose of an internal inspection is to detect external and internal cracks, buckles bulges and corrosion; leaks of welded, riveted and expanded joints; damaged setting. Each steam generator with all its fixtures is to be tested hydrostatically every four years by a pressure of 1.25 p_w, where p_w is the working pressure.

The inspections are conducted by representatives of the authority (e.g. the USSR Register Inspectorate) establishing the code of safe steam generator operation. The results of each inspection are entered into the steam generator service book. If there is evidence that the plant is in bad repair involving an operational hazard, it is withdrawn from service.

In the course of routine operations, entered into the service book are various particulars regarding the condition of the steam generator, fittings (relief valves, feed water valves, etc.), means of automatic control, the dates and durations of blowdowns, the date of the forthcoming inspection.

Steam generators are repaired in accordance with the schedule of preventive maintenance. The results of each repair are entered into the repair record along with the particulars of the plant condition before each cleaning (the depth of scale and sludge, the defects detected in the course of the repair).

REVIEW QUESTIONS

1. What purpose do steam generators serve?
2. What feature of design is used for classifying steam generators?
3. What is understood under the fittings of a steam generator?
4. What purposes do the main fittings of a steam generator serve?
5. What advantages does the automatic control of a steam generator offer?
6. What are the main features of steam generator operation?
7. When is a steam generator subjected to inspections?

Chapter XXIII

MARINE AUXILIARIES

126. Shipboard Pumps

Pumps are machines designed to transport various liquids. Used on board ships are a variety of types such as reciprocating, peripheral, jet, centrifugal, gear, screw and vane pumps.

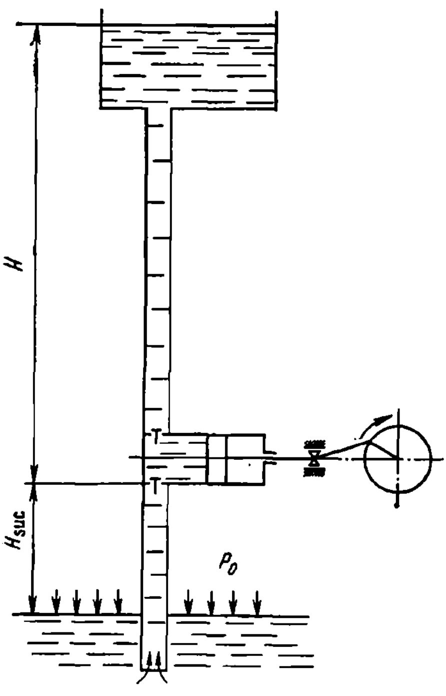


Fig. 147. Suction lift and delivery head of water pump

Pump performance is expressed in terms of suction lift, delivery head and capacity. A vacuum created in the suction pipe of a pump causes the liquid to enter the pump casing. The maximum suction lift H_{suc} (Fig. 147) is the height of water column which is set at balance by atmospheric pressure. Theoretically, it equals 10.33 m, but practically no pump can operate at a suction lift of over 6 m or 7 m as a maximum owing to the impossibility of creating a total vacuum in the suction pipe.

In performing its work of delivery, a pump causes the liquid to rise until the force produced by the pump is at balance with the pull of gravity exerted by a water column with a height H. The maximum height of delivery is termed delivery head and is expressed in metres of water column, or newtons per square metre.

Reciprocating pumps exist as single- and multi-cylinder units of the single- and double-acting type driven by either individual motors or crankshaft, a crank back-balance being used in this latter case. In a double-acting pump, delivery takes place from the spaces above and below the plunger. Therefore, the capacity of a double-acting pump is twice that of a single-acting one, other things (plung-

er speed, size) being equal. In shipboard application, reciprocating pumps are used to handle petroleum products, fresh and outboard water, etc. They are simple, reliable, efficient and capable of producing a significant suction lift. However, reciprocating pumps are heavy and bulky.

Centrifugal pumps are also simple, compact and highly efficient but cannot start operation unless being primed. This shortcoming is overcome in some installations by providing self-suction attachments. The drive may be obtained from an individual motor, though engine-driven pumps are also in use. The impeller may have a single inlet on one side or water may flow to it symmetrically from both sides. To obtain high heads, multistage pumps are used comprising a number of impellers connected in series on the same shaft. The total head is the head of each impeller multiplied by the number of impellers. They are used as fire and feed-water pumps.

When high capacity is essential in marine application, use is made of pumps having a number of impellers operating in parallel and delivering into a common manifold. Here the head delivered is the head of any impeller, while the capacity is the sum of all capacities. The multiflow type, as these pumps are referred to, is employed in cargo discharge systems of tankers, as bilge drainage pumps of salvage-and-rescue vessels and fire pumps of large ships, etc.

127. Fans, Shaft-Driven Generators and Air Compressors for Marine Application

Centrifugal marine fans operate in the same way as centrifugal water pumps and fall with reference to design into radial-flow and axial-flow types. In the former, the flow of air is from the centre to the periphery of the impeller and in the latter, the flow is along the shaft axis. Most of the marine fans are of the radial flow type made up mainly of welded parts.

To utilize the main engine power as fully as possible, and save engine room space, for no special diesel is required in this case, use is made of shaft-driven generators. A generator of this kind consists of a hollow frame giving support to the stators of the a.c. exciter and rotary converter. The armature of the generator is linked to the ship's shafting or the fore end of the crankshaft through a clutch and step-up gear. Shaft-driven generators are commonly used to carry the load on long passages with the main engine running at continuous rating. Automatic regulation is a feature of these generators.

Marine compressors are driven as a rule by electric motors, using V-belts. In this case, they are installed on the same base with the motor. However, most of the compressors in marine application dispense with V-belts, being directly geared to the motor. Ships with

a limited sailing range may employ engine-mounted compressors and exhaust-tapping valves.

To meet significant requirements in compressed air by means of a compact installation, the recourse is to double-stage compressors with plain or differential pistons, air intercoolers, oil and moisture traps. Free-piston diesel compressors also find application on board some ships.

The diesel compressor depicted in Fig. 148 is a high-pressure four-stage machine linked to a two-stroke engine with uniflow port scavenging so as to form a single unit contained in a common case. It

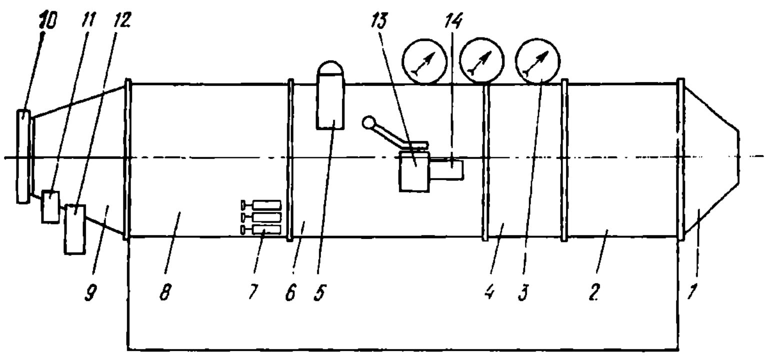


Fig. 148. General view of diesel compressor

is capable of delivering air under a pressure of 2 230 $\times$ 10^4 N/m^2 (230 kgf/cm^2). Diesel engine *6* is the midmost constituent flanked by the second and third compressing stages *8*, *9*, *10* at the right-hand side, and by the first and fourth compressing stages *1*, *2* at the left-hand side. The first stage doubles as the scavenging pump. The fuel-injection pump *5* with a filter, starting air distribution valve *13*, manual starting valve *14*, the water pump and lubricator are fitted to the housing from the outside. Accommodated in the upper part of intermediate case *4* is an automatic starting aid topped by panel *3* mounted on which are the pressure gauges. Air intercoolers serving all four stages are provided at the bottom of the unit. A fourth-stage delivery valve *12* together with relief valve *11* are built into the line downstream of the fourth stage air intercooler. The blow down valves of the four stages are carried by bracket *7*.

At the heart of the diesel compressor are two groups of pistons reciprocating in opposite directions. Each group consists of the engine pistons directly linked to the compressor pistons. A synchro-

nization mechanism keeps the strokes at the same length, providing for stable performance. The gears of this mechanism set into motion the fuel-injection and water pumps, and the lubricator.

REVIEW QUESTIONS

1. What types of pumps are installed in ships?
2. What purposes do marine pumps serve?
3. What influences the suction lift of a pump?
4. What influences the delivery head of a pump?
5. What are the advantages of centrifugal water pumps? Contrast the advantages and drawbacks of centrifugal pumps and those of reciprocating ones.
6. Why are shaft-driven generators used on board ships?

BIBLIOGRAPHY

1. VANSCHEID, V. A., "Sudovye dvigateli vnutrennego sgoraniya" (Marine Internal Combustion Engines), Sudpromgiz, Leningrad, 1957.
2. VESHKELSKY, S. A. and SVETLICHNY, M. N., "Montazh, expluatatsiya i remont dvigatelei vnutrennego sgorania" (Installation, Operation and Repair of Internal Combustion Engines), Mashinostroenie, Moscow-Leningrad, 1966.
3. VOZNITSKY, I. V. and CHERNYAVSKY, N. G., "Sudovye dvigateli vnutrennego sgoraniya" (Marine Internal Combustion Engines), Transport, Moscow, 1974.
4. KANE, A. B., "Reversivnye ustroistva sudovykh diselei" (Reversing Gears of Marine Diesels), Sudostroenie, Leningrad, 1965.
5. KANE, A. B., "Bor'ba s vzryvami v kartere sudovykh diselei" (Prevention of Crankcase Explosions in Marine Diesels), Sudostroenie, Leningrad, 1969.
6. KANE, A. B., "Sudovye dvigateli vnutrennego sgoraniya" (Marine Internal Combustion Engines), Sudostroenie, Leningrad, 1973.
7. MASLOV, V. V., "Sudovye sistemy malooborotnykh diselei" (Slow-Speed Diesel Auxiliaries), Sudostroenie, Leningrad, 1968.
8. POMUKHIN, V. P., "Diselnye ustanovki, mekhanismy i oborudovanie promyslovykh suden" (Diesel Engines and Auxiliaries of Fishing Vessels), Sudostroenie, Leningrad, 1974.

INDEX